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The Marine Engineer

AND NAVAL ARCHITECT.

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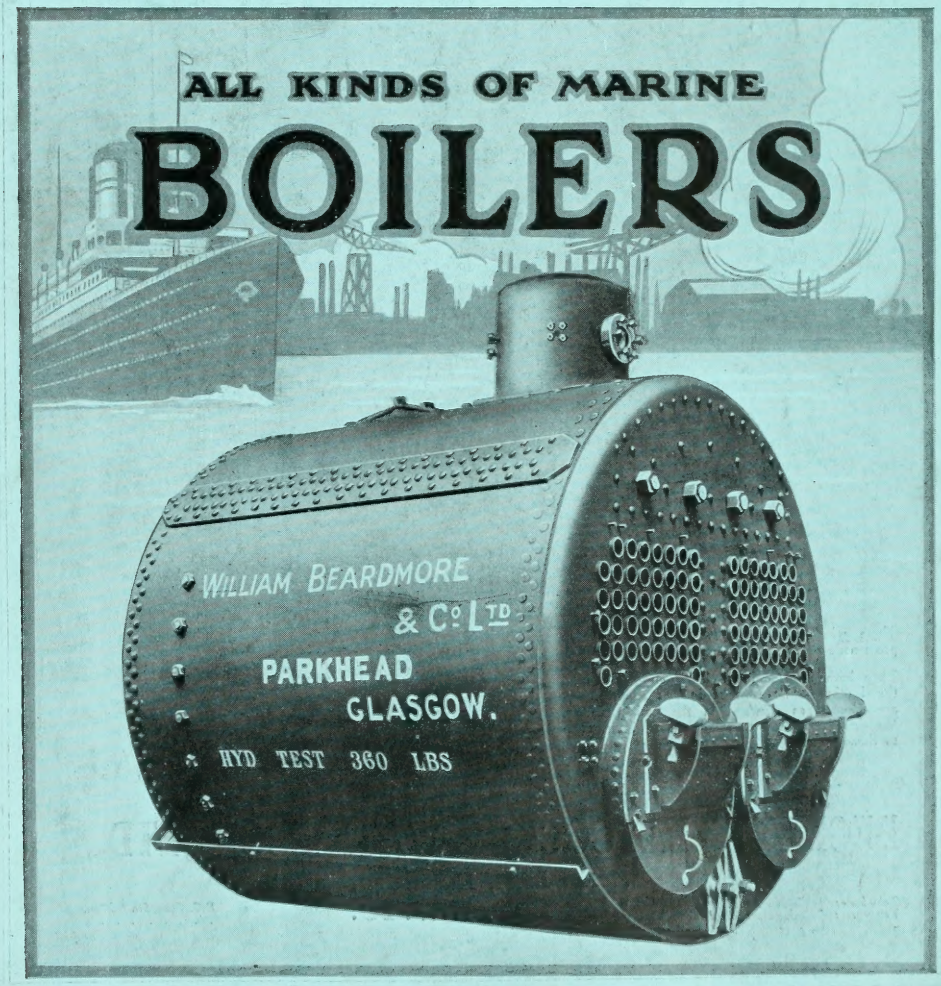

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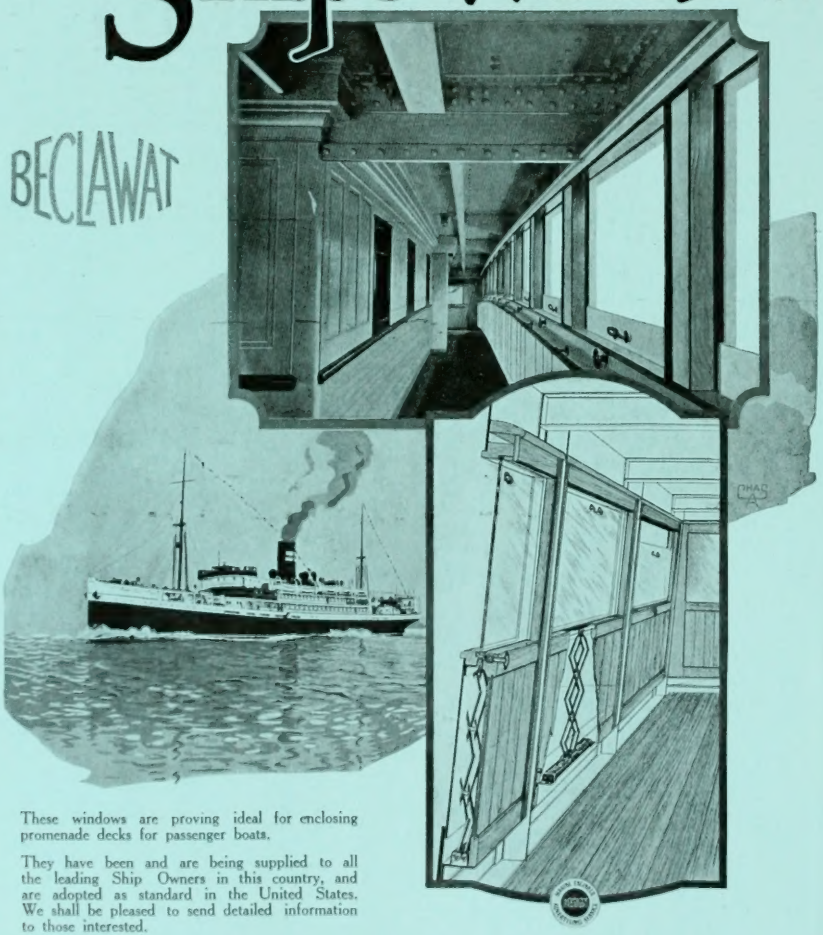

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The Marine Engineer
AND NAVAL ARCHITECT.

A Practical Journal for all Concerned in the Design Construction and Maintenance of

MECHANICALLY PROPELLED SHIPS — MARINE ENGINEERING WORKS
DIESEL AND OTHER OIL ENGINES — WELDING & MECHANICAL REPAIRS
AUXILIARIES & REFRIGERATION — CARGO HANDLING APPLIANCES

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The Editor will be glad to consider original articles and paragraphs on all subjects indicated by the title. Short practical articles based on the sender's own experience and illustrated by photographs or drawings are desired.

VOL. XLVI. No. 550. JULY, 1923.

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The death has occurred of Mr. William Shreeve Doran, President of the Alberger Pump & Condenser Company, New York.

Mr. George F. Murphy has been placed in charge of the New York office of the Heine Boiler Company. He was formerly Pittsburg Manager for the same company.

Mr. J. C. Cliffguard, who is Assistant Manager of the construction and repair department of the Standard Oil Company, is now in Europe for the purpose of investigating different types of Diesel engines.

Mr. James McNaught, of the firm of Esplen, Sons & McNaught, naval architects and marine engineers, is also at the present time reporting upon various European types of Diesel engines for the Standard Oil Company.

The death has occurred, at Newcastle, of Mr. David A. Sheret, Consulting Engineer, formerly Lloyd's Surveyor, Chief Engineer for John Mowlem & Co., Engineer-in-Charge of the Marine Department of Southern Nigeria, and Engineer of the Royal Indian Marine.

The death has occurred at Gibraltar of Engineer-Captain Frederick Barter, O.B.E., R.N., Chief Engineer of H.M. Dockyard, Gibraltar.

Lieut.-Colonel Charles Faunce Hitchins, D.S.O., M.I.Mech.E., has been appointed General Manager of the Elswick Works of Sir W. G. Armstrong, Whitworth & Co. Ltd. For some time he has been Manager of Agricultural & General Engineers Ltd.

Mr. E. W. Petter, the newly-elected President of the British Engineers' Association, is Chairman of Petters Limited, Yeovil, and managing director of Vickers-Petters, Ipswich; he is a recognised authority on internal-combustion engines.

Mr. Sydney Evershed, who has recently retired from the position of Managing Director of Evershed & Vignoles Ltd., has now accepted the office of Chairman, and will in addition act as Consultant to the Company. Mr. E. B. Vignoles and Mr. A. Vines remain managing directors of the company.

The late Mr. Robert Richardson, of Liverpool, Superintendent Engineer of the Harrison Line, and a native of Berwick-upon-Tweed, left gross estate of the value of £75,161, with net personalty £74,709. Among other bequests he left £1,000 to the Liverpool Seamen's Orphanage.

Mr. W. J. Willett Bruce, Superintendent-Engineer of the White Star Line, has been promoted to the honorary rank of Engineer-Captain in the Royal Naval Reserve. Mr. Willett Bruce has been Superintendent-Engineer of the company since 1904, and in 1911 was President of the Liverpool Engineering Society. In June of last year the Liverpool University conferred upon him the honorary degree of Master of Engineering.

The Lord President of the Council of the Department of Scientific and Industrial Research has accepted with regret the resignation of Sir George Beilby, LL.D., F.R.S., after nearly seven years' voluntary service as Director of Fuel Research and Chairman of the Fuel Research Board. He has appointed Mr. C. H. Lander, D.Sc., M.I.Mech.E., A.M.Inst.C.E., to be Director of Fuel Research, and Sir Richard Threlfall, K.B.E., F.R.S., a present member of the Board, to be Chairman. Sir George Beilby retains his membership of the Advisory Council of the Department, and has consented to act as Honorary Adviser to the Board.

Major Gilbert S. Szlumper has been appointed by the Board of Directors of the Southern Railway Company to be Docks and Marine Manager to the whole of the group which comprises the London & South Western Railway Company, London Brighton & South Coast Railway, and South Eastern & Chatham Railway Company, and from July 1, in addition to Southampton docks, he will also assume command of the ports of Newhaven, Folkestone and Dover. Major Szlumper's responsibilities will include the supervision of 44 vessels of various types, embracing the cross-channel boats running between Southampton and France, Southampton and the Channel Islands, Newhaven and Dieppe, Dover and Calais, and Folkestone and Boulogne.

The death has occurred of Mr. John Fadden, Engineer-Surveyor in Chief and Principal Examiner of Engineers to the Australian Commonwealth Navigation service. The late Mr. John Fadden, on the completion of his apprenticeship at the St. Rollax locomotive works, Glasgow, decided to learn marine engineering and transferred his activities to J. & G. Thompson, Clydebank (now John Brown & Co. Ltd.). After gaining the necessary experience of a marine engineer he shipped as third engineer on the ss. *Carrie*. Later he settled down in Australia, and took up an appointment as engineer of the Sydney paper mills. Having passed as a marine engineer surveyor, he was appointed Senior Engineer-Surveyor to the Victorian Marine Board in 1915. In May, 1921, he was transferred to the Commonwealth Navigation Service, and held the position of Engineer Surveyor-in-Chief and Principal Examiner of Engineers until his death.

EDITORIAL.

SEA TRAINING FOR ENGINEERING STUDENTS.

IT is interesting to hear of a scheme such as that to be carried out by the White Star Line through Engineer-Captain W. J. Willett Bruce, R.N.R., whereby engineering students from the Liverpool University are enabled to get practical experience at sea during the summer vacation in the vessels of that company. The university-trained man plays an important part in the engineering industry, but he is often condemned at the outset of his career as being too theoretical and of failing to be completely *au fait* with the more practical side of his profession. This depends, of course, largely upon the individual and much upon his scheme of training. The man who is on the engineering side at his school, for instance, and who then pursues his absorption of knowledge at a 'varsity, straight away, is unquestionably more theoretical than the man who goes from school straight into the shops, where all his "rough edges" are removed and he assimilates a more generous outlook on life, and then proceeds to the 'varsity, where, incidentally, he should be able to absorb a good deal of pleasure as well as of knowledge before entering upon his profession seriously. In either case, however, as far as the practical sea-going working of marine engineers is concerned the newly-fledged graduate will find himself lacking. The proportion of shipowners who will take a marine engineer, or a naval architect student, during his vacation and give him the opportunity of obtaining some experience which may well prove invaluable in after life, is not great, although, we would emphasise, by no means non-existent. It has, we know, been suggested in various quarters that the leading students in each year's examination at some of the 'varsities should be given privileges such as those outlined above, but as far as we are aware nothing has come of the suggestion. In any case such an idea is wrong, fundamentally, because such privileges should definitely not be accorded to the most brilliant examination students (who are in some cases the most brilliant failures in after life). Any such scheme should be open, without prejudice, to all students except to those whose obvious lack of interest, and unfortunately 'varsities are peopled with many such, renders them unfit to receive the benefits accruing from the idea. The important point is that a definite impetus has been given to such schemes by the White Star Line in conjunction with Liverpool University, and it will be interesting to see how long a time will elapse before the other engineering universities are taken under the wing of kindly-disposed steamship companies.

ELECTRIC DRIVE FOR WARSHIPS.

WE publish on page 262 an abstract from a paper read before the American Society of Engineers dealing with the propelling machinery of the battleship *Maryland*. The American Navy, it will be recalled, has championed the cause of the electric drive for many years, the first exponent being the fleet collier *Jupiter*, converted recently into an aircraft carrier, as the *Langley*, similar somewhat externally to our own *Argus*. Prominence was given to American opinion on the subject by Mr. W. L. R. Emmet in his very able paper at this year's spring meetings of the Institution of Naval Architects, at which paper also the feeling of our own Admiralty was forcibly expressed by Engineer-Vice-Admiral Dixon.

For fighting ships, as distinct from merchant ships, the advantages of the electric drive are mainly strategic, and they were rather neatly summed up by Rear-Admiral C. W. Dyson, U.S.N., in an article entitled "Development of Machinery in the United States Navy during the Past Ten Years," which appeared in the Journal of the American Society of Naval Engineers for May, 1917. Some of the principal benefits he listed were:—1. Greatly increased torpedo protection for ships. 2. Greater flexibility in machinery arrangement. 3. Better and wider separation of important units. 4. Minimum lengths and diameters of steam pipes, and fewer bulkheads pierced by steam and feed piping, with consequent reduced heating of the vessel from steam pipes. 5. Better centralisation of power. 6. Reduced engine-room complement. 7. Elimination of danger from fractures of piping due to shells striking protective deck. 8. Better maintenance of economy through a wide range of power. 9. Maximum reduction in length of shafting.

In the current number of the Journal of the American Society of Engineers, Lieut.-Commander A. M. Charlton, U.S.N., in his very interesting article entitled "Electric Drive on Battleships," arrives at the conclusion that "the application of the induction motor for the propulsion of ships presents such a radical departure from the time honoured reciprocating engine and the later development of the turbine, that most naval officers are bewildered by the apparent complexity of the machinery layout in an electric drive ship." He thus virtually takes the standpoint that conservatism is the root reason for much of the criticism directed at the electric drive, and in this possibly there is a certain amount of truth; nevertheless, a perusal of his article certainly leaves in its wake the impression that the American Navy, at any rate, has every reason to be satisfied with its choice. The system was to have been extended also to American battle-cruisers we believe, but in any such vessels the test of reliability will necessarily be much more severe on such large installations, both on account of the higher speed required (30 knots may be slow even for the battle-cruiser of the future), and also because the battle-cruiser is essentially a more mobile unit than the battleship proper.

Coming now to smaller naval craft, we find that America has also given the electric drive a trial in some of her latest revenue cruisers. These latter are a type of ship which has no exact prototype in the British Navy unless, of course, we consider the "Flower" class and other sloops which have survived the post-war economy schemes. Two United States Revenue cruisers, it will be remembered, maintain the International Ice Patrol off the Grand Banks of Newfoundland during certain seasons of the year, and vessels equipped with electric drive have been placed on this station recently and it is stated that their manœuvrability has proved extremely valuable which we can well believe if the vessels have had much navigation to carry out in infested waters. Naval auxiliaries were amongst the first vessels to be fitted out with electric drive, as has been mentioned, but it is interesting to recall that patents were taken out for the use of electric motors for electric propulsion as far back, comparatively, as 1893. The General Electric Company equipped two fire floats for the city of Chicago in 1908 in which turbo-generators furnished power for the propelling motors; these in turn were mounted directly on to the propeller shaft. The same company later submitted plans for the electric propulsion of the battleships *Wyoming* and *Pennsylvania*. The Navy Department, however, were not quite ready to accept this proposal at either time, but the building of three sister vessels *Jupiter*, *Cyclops* and *Neptune*—fleet colliers—gave to the electric drive an opportunity to be tried against reciprocating engines and geared turbines in the *Cyclops* and *Neptune* respectively. The resulting performances of the electrically-propelled vessel, were very satisfactory and as Lieut.-Commander Charlton states, in the article to which we have referred above:—"The performance of the *Jupiter* set at rest all fears as to the reliability of electric machinery for ship propulsion. At the same time the economy of her installation over that of her two sister ships was quite marked."

VACUUM IN MARINE ENGINES.

THE highest vacuum to which the reciprocating engine should work for economy in fuel is a question on which differences in opinion exist among engineers. Unlike the steam turbine, limitations in dimensions of low-pressure cylinders prevent the full expansion of the steam to the back pressure line being obtained, nor is it desirable, since frictional resistance of these engines would make full expansions useless as regards the doing of useful work. To utilise a lower condenser pressure ample area must be supplied in exhaust ports, passages and eduction pipe, to allow of the steam leaving the engine at a moderate velocity, otherwise the potential benefits of the higher vacuum will be partially or wholly thrown away in increased leaving losses. Owing to these reasons the gain in economy of steam compared with that obtained in steam turbine work is small, and a limit is soon reached in which the increased expenditure of steam by the auxiliaries in creating the higher vacuum is balanced by the heat equivalent of the extra work done, due to reduction of back pressure on engine piston. The general belief is that in reciprocating engine practice no appreciable gain is effected by working with a vacuum higher than 26½ to 27 in., although experiments in individual cases have shown that higher values could be used with advantage. Each particular case, however, depends on its own merits, and the most economical condenser pressure can only be determined by observations made under service conditions.

In a paper read before the North-East Coast Institution of Engineers and Shipbuilders in November, 1910, on this subject, Mr. D. B. Morison gave some interesting data obtained from experiments made on engines of both the triple and quadruple expansion type, which indicated the possibility of improvement in economy by working with vacua up to 28 in.; these experiments, although inconclusive, show, as the author stated, that provided the low-pressure cylinder exhaust ports and passages are of ample area there seems no reason why higher vacua with these engines could not be economically employed. With the modern steam plant on board ship the deleterious effect of the low hot-well temperature, associated with high vaccum, on the economical generation of steam in the boilers, is compensated for by heating of the feed with the exhaust steam from the numerous auxiliary engines.

With turbine engines having dimensions suitable for their output no great difficulty is experienced in utilising the highest vacuum which it is possible to create in the modern condensing plant; care must be taken, however, in designing the exhaust blading that sufficient area is provided to deal economically with the large volume of steam at the lower pressure. In special cases, however, such as modern torpedo craft and other high-speed vessels, in which large outputs are developed by units of limited dimensions, difficulty is experienced in providing the necessary area through the blading to cope with the large quantity of steam passing, and in these cases the turbine becomes similarly circumstanced to the reciprocating type. Owing to restricted area the steam is unable to expand to the condenser pressure, and consequently increased condenser vacuum in these cases would constitute a loss instead of a gain in economy. To utilise the high vacuum economically, sufficient exhaust area through the blades must be provided with blade angles of moderate amount, otherwise the losses to the exhaust will become prohibitive. The angles of discharge should not exceed from 45 to 55 deg. in turbines of both the reaction and impulse type, as with greater angles the blades become very insufficient and their resistance to bending action is small. Wherever possible a moderate diameter of turbine at exhaust end should be chosen, having blades of nominal curvature, in order to deal effectively with the exhaust steam. The main eduction pipe should be as large in area as practicable to minimise the friction losses in the exhaust, the limiting speed generally adopted in marine practice with high-speed turbine work being about 400 ft. per second, which represents a loss of about 1 per cent. of the total available heat to the turbine due to velocity alone, while the friction losses may be from 1½ to 3 per cent., depending on length of pipe.

From the "Marine Engineer," July, 1879.

THE NEW TRANSATLANTIC CABLE.—The cable steamer *Faraday** left Millwall at an early hour on June 23, and during the forenoon was safely moored abreast of Messrs. Siemen's works at Charlton. The *Faraday* is a powerful steamer of 4,900 tons, built at Newcastle in 1874 for Messrs. Siemens, and designed especially for laying cables. In the course of the afternoon she was surrounded by barges; those on the outside supplying her bunkers with coal, and those next to shore being employed in passing the cable into the steamer's tanks. The new company, the official style of which is "La Compagnie Francaise du Télégraphe de Paris a New York." has opened offices in London for the transaction of executive business at 100 and 101, Gresham House, Old Broad Street. The *Faraday*, having on board the shore end and the Brest-Scilly section of the new cable, was cleared at the London Custom House on Tuesday night for New York, and sailed for Brest the following morning.

"ORIENT."—On June 5, J. Elder & Co. launched from their shipbuilding yard at Fairfield one of the largest steamers ever built on the Clyde. The ship bears the name of the company for whom she has been constructed, viz., the Orient Steam Navigation Company (Limited), and is intended to augment the fleet trading between this country and Australia. Her dimensions are:—Length, 460 ft.; breadth, 46 ft. 6 in.; depth, 37 ft. 8 in.; displacement at load draught, 9,500. Her engines, presently being finished by Messrs. Elder, are of the three-cylinder compound surface-condensing class, the high-pressure being 60 in. diam., and the two low-pressure 85 in., having a 5 ft. stroke. They are fitted with Brown Brothers' hydraulic and steam starting and reversing appliances, and Gwynne's patent centrifugal pumps.

THE LATE CUNARD LINER "SCOTIA."—Few would recognise in the large twin-screw steamer which left the Mersey on Thursday last the once famous Cunard liner *Scotia*, the last of the great paddle-steamers built for the Atlantic trade, and which was for years looked upon as the fastest and favourite vessel on the line between Liverpool and New York. The *Scotia* was built in 1862, when, with the exception of the *Great Eastern*, she was probably the largest mail steamer afloat, being about 400 ft. long over all, 47 ft. 8 in. beam, and 4,050 tons builder's measurement, and fitted with a pair of side-lever engines of 1,000 h.p. The introduction of screw steamers fitted with compound engines for the Atlantic and other ocean voyages has of late years entirely superseded the paddle-steamers, and a few years back the *Scotia* was withdrawn from the Cunard Company's sailing list, and was subsequently purchased by the Telegraph Construction & Maintenance Company to be employed in their cable-laying operations. To fit her for this work very extensive alterations were, of course, necessary, the execution of which the company entrusted to Laird Brothers, and during her stay at the Birkenhead Iron Works the *Scotia* had been stripped of her funnels, machinery, paddle-wheels, deckhouses, &c.; she has also been raised by the addition of a spar deck and altered about the after end to prepare her for twin-screws, she has been fitted with new compound engines and also provided with three immense cylindrical tanks in which to stow the electric cable, as well as with most elaborate and approved steam machinery for paying out and hauling in; also steam capstan, steam steering gear, winches, &c. The new engines are two distinct sets on the compound system, with inverted cylinders 38 and 66 in. diam., and 3 ft. 9 in. stroke.

* See *The Marine Engineer and Naval Architect*, February, 1923.

BOOKS FOR THE MARINE ENGINEER AND NAVAL ARCHITECT.

Brief Reviews of Recent Publications.

Jordan's Tabulated Weights of Iron and Steel.—Edited by Lloyd's Register of Shipping, London: Published for Lloyd's Register of Shipping by E. & F. N. Spon, Ltd., 57, Haymarket, S.W.1. $5\frac{1}{4}$ in. × $3\frac{1}{4}$ in. × $\frac{5}{8}$ in. 316 pp. Price 15s. net.

This is the eighth edition of this useful little book, revised in the light of the new Rules and Regulations of this Society, issued in July last; it includes also the new Sections drawn up by the British Engineering Standards Association in 1920.

The Electrical Handling of Materials.—Vol. IV. By H. H. Broughton, M.I.Mech.E. London: Ernest Benn Ltd., 8, Bouverie Street, E.C.4. $8\frac{3}{4} \times 10 \times 1$ in. 334 pp. Price 50s. net.

Volume four of this manual on the design, construction and application of cranes, conveyors, hoists and elevators, deals with machinery and methods. The book is well produced and profusely illustrated and of special interest is Chapter V, which deals with the bulk-handling of cargoes in America. Both the Great Lakes (upon which most of the bulk cargo work is done) and the types of vessels which navigate them are, as it were, a chapter apart in the work of ship operation, and the plant which has been erected to deal with the enormous quantities of cargo handled is amongst the finest in the world. There is also a very complete chapter dealing with the handling and storing of grain. These are but two typical selections from the work, even the most cursory glance at which shows that the author has prepared his manuscript and its accompanying diagrams with considerable care. We can recommend the book to engineers with confidence.

Marine Engineer's Drawing Book for Board of Trade Examinations. By W. C. Macgibbon, M.I.Mech.E. Glasgow: James Munro & Co. Limited, 35, Washington Street. 11 in. × $8\frac{1}{2}$ in. × $\frac{5}{16}$ in. 132 pp. Price 8s. 6d. net.

This book has been rearranged owing to alterations in the examination of engineers for Board of Trade Certificates, which the author commends in his preface. The letterpress commences with a short descriptive article entitled "Drawing Hints for Beginners." This is followed by a section dealing with the proportion of parts, which is, if anything, formula-ridden, although concise. The author has included a section dealing with stresses in working parts, compressing much that is useful into a small space. Cylindrical vessels and their stresses occupy one complete page, and the rest of the book comprises sketches and explanatory matter, a helpful feature in these being the shaded perspective sketches accompanying the explanatory notes. The volume is completed by two pages of notes for engineers, which form a pot pourri of formulæ likely to be required in practice, and two detail tables of Whitworth standard bolts and nuts. The book is, according to the title, a pictorial drawing book for marine engineers, but we think that the author could with advantage have qualified his title by mentioning that it dealt mainly, if not entirely, with steam reciprocating engine practice, which at the present day is but a branch of marine engineering. The 132 pages contain, however, much that is useful in this particular branch and should prove of value to students.

Sous-Marins, Torpilles et Mines. By M. Laubeuf and Henri Stroh. Paris: Libraire J. B. Baillière et Fils, 19, Rue Hautefeuille. $6\frac{1}{4}$ in. × $9\frac{1}{4}$ in. × $1\frac{3}{4}$ in. 800 pages. Price 60 fr.

This book forms one of the Encyclopædia of Applied Mechanics, edited by M. L. Lecornu, under the patronage of the Society of Civil Engineers in France and of the Society for the Encouragement of National Industry. Submarines, perhaps more than any other type of war vessel, were the subject of vast research and of considerable development during the war, especially in Germany, as everyone knows, but it should be remembered that German development was of the meteoric type, since she did very little towards submarine perfection before the war; it is interesting, in fact, to note, as the authors themselves state, that in 1901 Admiral von Tirpitz declared in the Reichstag that Germany, by reason of the nature of her coast lines and of her geographical situation, had no need of submarines, which he regarded as purely defensive weapons! France has always been to the fore in submarine development, and her work in connection with the development of the double-hulled *submersible*, fostered largely by M. Laubeuf, as contrasted with the single-hulled *submarine* with external ballast tanks, as developed in the British Navy, is well known. The book we have under review is the second work on submarine craft which has appeared in Europe within the last few months. The other* one we have in mind, which was reviewed in these columns in January last, was a German exposition, and it dealt with the subject very thoroughly. This book, however, is more than an exposition: it is almost an encyclopædia in itself, and it can be read, moreover, with the feeling that the authors are experts in the subject upon which they are writing, and upon which they state that although immense progress has been made there still remains much more to be done. Part I deals with submarines and submersibles, and Chapter I is historical; Chapter II deals with the main types of submarines and submersibles and points connected with their sub-division; it also treats very thoroughly of the strength of submarine hulls, and includes a section dealing with the theory of M. Doyére. Chapter III deals with stability, and then follow sections dealing with main and auxiliary machinery, diving gear, armament, navigating gear, &c. There is a chapter which gives a resumé of submarine construction since 1914, and which devotes, as might be expected, several pages to Germany. The second section of the book deals very thoroughly with torpedoes and mines, and some plates of submarines are included at the end. The book is a bulky one, which contains much interesting and valuable information, although it cannot be said to enunciate many new principles.

TRADE PUBLICATIONS.

The Beldam Packing & Rubber Co. Ltd., 29, Gracechurch Street, E.C.3, have issued a small illustrated book descriptive of their well known packing and other products.

The Aster Engineering Company, Wembley, Middlesex, have issued a folder containing a selection of coloured illustrations of various "Aster" engines and combined plants of standard and special design.

Chambers, Scott & Co. Limited, Dalziel Engineering Works, Motherwell, have produced an illustrated booklet dealing with their electric winches and deck machinery, and containing a reprint from a supplement to the *Engineer*, June 21, 1907, entitled "The Economical Handling of Cargo, Boats, Baggage, &c., by Electric Motors."

* "Die Unterseeboote der Germaniawerft," by Dr. Ing. H. Techel. Julius Springer, Berlin.

LETTERS TO THE EDITOR.

THE NOMENCLATURE OF OIL ENGINES.

TO THE EDITOR OF
THE MARINE ENGINEER AND NAVAL ARCHITECT.

Your issue of February last contains a copy of the Oil Engine Nomenclature Committee's Report issued from the Institution of Mechanical Engineers, London.

Clause 3 in that report states :—

> " In the development of modern oil engines, departures have been made both from the ' Akroyd ' and from the ' Diesel ' original principles, &c., &c., &c."

In justice to myself, may I ask the courtesy if you will grant space in *The Marine Engineer and Naval Architect* for my explanation and challenge to the oil engine world ?

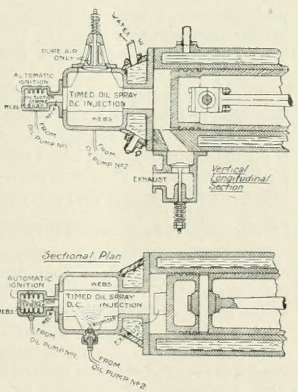
I maintain that for general purposes all types of heavy oil engines may be classed under true and simple terms which more correctly describe the heavy oil engine than " Diesel " and " semi-Diesel." Therefore, why not discard these terms ?

Established practice does not warrant a continuance of an injustice. The coming race of engineering students and marine steam engineers must be taught the correct origin of the cycles on which heavy oil engines work. It was as much for the credit of the British nation as myself that I persevered for a period of over six years to prove my claim, *i.e.* to be the pioneer inventor of the *two* oil engine cycles from which has been developed the heavy oil engine industry throughout the world.

Now it is established that all automatic ignition heavy oil engines work according to one or the other of my Akroyd automatic ignition heavy oil engine cycles of 1890—or a combination of them—*see* the accompanying sketches.

Reference to the above combination of Akroyd vaporisers was made in my letter to the editor of the *Engineer*, published in the *Engineer*, November 4, 1921.

The Akroyd is the original fundamental principle on which all heavy oil engines work ; also it is the original combination principle—for, for all-round purposes it is possible to classify all heavy oil engines on a compression pressure basis.



Sketches showing Engine Working on the Akroyd Cycle of 1890.

Class No. 1, low and medium compression Akroyd hot bulb (automatic ignition).

Class No. 2, high compression Akroyd cold bulb (automatic ignition).

These terms express the original and the true historic evolution of the modern heavy oil engine.

The Hornsby and the Petter automatic ignition heavy oil engines are examples of Class 1 engine, whilst the Ruston, Beardmore, Crossley, Vickers automatic ignition heavy oil engines are examples of Class 2 engines. If manufacturers of the oil engines in question would get together and determine the limit compression pressure for practical working for the Class 1 engine, then the Class 2 engine provides for all other compression pressures. Also it would be advisable that each manufacturer should describe and illustrate the special patented device used to " spray " or " atomize " the liquid fuel for combustion, then every engineer who has been instrumental in this branch of engineering could have his share of honour in the development of the modern oil engine.

It is often stated in engineering journals, textbooks, institution " papers " and the like, that the " Hornsby-Akroyd," is a low compression oil engine, and the engine cycle is obscure. Such a statement is wrong and misleading. The patent Specification No. 15994, October 8, 1890, provides that the oil may be injected on the compression stroke, *see* page 1, lines 46, 47 and 48.

> " The engine shown in the drawing is designed to have the liquid hydrocarbon injected into the explosion chamber and formed into spray *before* impinging upon the heated walls of the said chamber," &c., &c. Page 2, lines 1, 2 and 3.
>
> " The injection may be so timed as to occur at the beginning or any portion of the suction stroke, or during the *compression stroke*."

May I ask the favour that further comment on my early Akroyd automatic ignition heavy oil engine cycles may refer to my work as whole, as described in the specifications and drawings of patents No. 7146, May 8, 1890 ; No. 15994, October 8, 1890 ; Akroyd water or fuel oil-cooled vaporiser, No. 3909, February 29, 1892.

Those were the basic principles I laid down 30 years ago to produce eventually an efficient and reliable oil engine. Akroyd engines built according to the above-mentioned patents were sold and in daily use in England, France and Germany before Dr. Diesel took out his *first* patent for oil engines in Great Britain.

In order to prove the above statement, that the modern automatic ignition heavy oil engines are a natural evolution of my Akroyd engines of 1890, I am prepared to build an Akroyd automatic ignition heavy oil engine according to the specifications and drawings of patents No. 7146, May 8, 1890 ; No. 15994, October 8, 1890 ; and Akroyd water or fuel oil-cooled vaporiser patent No. 3909, February 29, 1892 ; such an Akroyd engine to work on a compression pressure starting from 45 lb. per square inch (hot bulb, class 1), rising in stages up to 400 lb. pressure per square inch (cold bulb, cold starting, class 2), as set out in Mr. Liven's paper read before the Members and Council of the Institution of Mechanical Engineers at Lincoln, July, 1920.

That Akroyd oil engine to use oil as fuel from the start (specific gravity ·854, " Russaline " or " Scotch Shale Oil "), similar to that used in the Akroyd automatic ignition oil engine which Professor Robinson tested for me at Bletchley Iron Works, Bletchley, Bucks, February, 1891, the density of the oil to be increased as compression pressure is increased and the cooled surface of the combustion chamber extended until the high compression pressure stage, 400 lb. per square inch, is reached, when the surface of the combustion chamber will have been entirely cooled (Class 2, cold bulb, cold starting oil engine).

Apropos to the dual or compound Akroyd vaporisers, " two oil injection method." It may interest your readers to know that in 1905 Messrs. Crossley Brothers, of Manchester, altered one of their Crossley oil engines, 20 b.h.p., to work according to my " two oil injection method " (patent No. 28045, December 21, 1904).

Mr. W. Le P. Webb, Messrs. Crossley Brothers' oil engine expert, tested that 20 b.h.p. engine and his report on it was very satisfactory ; in fact, I submitted it for perusal to the members of the Oil Engine Nomenclature Committee during their term of office at the Institution of Mechanical Engineers, London.

HERBERT AKROYD STUART.

Akroydon,
Claremont, W. Australia.

SOME NOTES ON ADJUSTMENTS AND REFITS TO MARINE RECIPROCATING MACHINERY.

Crank Head Bearing Adjustment—Piston Ring Fitting—Slide Valve Setting—Friction Test for Shafting.

The following notes are of a strictly practical nature, and they deal with some important and interesting adjustments and refits which have to be carried out from time to time on vessels propelled by steam reciprocating machinery. The notes have been made as concise and brief as possible and all unnecessary detail has been omitted.

No. 1.—Adjustment of Crank head bearing.

Notice the exact amount the nuts are screwed up from the scale and slack them away; then screw in the eyebolts, and lower the bottom brass into the crank pit. Turn up the crank shaft and sling the piston by means of a bar under crosshead; then turn down crank pin clear of the upper brass.

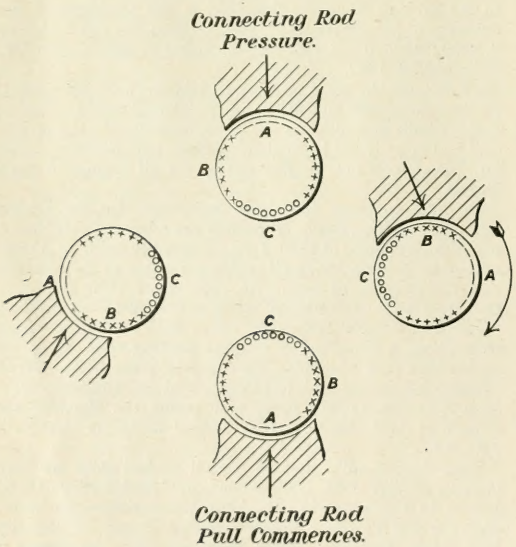


Diagram showing Alteration of Direction of Push of Connecting Rod in One Revolution.

Carefully examine the brasses to see that all lubricating ways and oil-ways are correct. Examine the pin for any indentations or water marks. Test the pin for trueness by callipers and also to ascertain if it is parallel; if these points are satisfactory, try further with marking. If everything is correct, connect up, and with spanner drive hard home. Take the "hard up" mark carefully, calliper the distance the brasses are open, and then fit liners, their actual thickness depending upon the working clearance it is desired to allow the pin. In first-class practice in a large ship, thirty thousandths is usually allowed.* Lower away the brasses, put the liners in place and take leads (usually two are taken round the pin and one fore and aft). Then drive hard up on the liners and take marks. After this, lower away and examine the leads, and if these are compressed satisfactorily, take up to the marks, and see if the brasses are free to move along pin by means of a crowbar. If they are quite free, fit them up and lock them. If a micrometer pin gauge is carried on board, try the pin with this in preference to callipers. It is interesting to note that the wear of a crank pin is practically only on one-half of its circumference, as will be seen from the diagram. The diagram shows how, in the course of a revolution, the push of the connecting rod is on the half of the pin A B C through the down stroke, and the pull is also on A B C through the up stroke, wearing A B C. Such wear should be measured and allowed for during adjustment. If a crank pin is badly marked and requires filing, a good plan is to make a wooden lap, halved, and bored to the diameter of pin. Emery powder may then be used and the pin finally trued with the lap.

* A useful rule is to allow one and a-half thousandths for each inch diameter of pin.

No. 2.—Fitting a piston ring for a large cylinder.

We will, for the sake of definiteness, consider the low pressure piston ring. Before the ring is tried, the tongue piece must be examined carefully to see that it is correct. The ring is then put into the cylinder (not on the piston), and gauged by feelers all round at top, bottom and centre positions. It is then lowered on to the piston and coach springs are put in, and the ring carefully gauged all round to see that the springs are doing their work. The junk is then temporarily put on, and the engines turned to top, bottom and centre of stroke; at each position the ring must be gauged all round. If this is correct, lift the junk and take ring out. Now place four pieces of lead tube with tapered edges, and a little deeper than the ring, vertically on the body of piston. The junk is carefully lowered on to the lead tubes, nuts are put on and tightened up. Next the junk is lifted, and the depth of the leads is measured. If these are a little deeper than the ring, about forty thousandth is a good clearance, then lower the junk, tighten up and place on the keep ring. Finally turn the engines through three or four revolutions to see that everything is correct.

No. 3.—Sighting through for main shafting in a new ship.

Two centres are taken from the drawings, one on the forward engine room bulkhead, at given distance below water level and from centre line bulkhead, and the other as near as possible to the centre of after end of A bracket. The former is marked with a $\frac{1}{32}$-in. hole, through a small index plate. A straight line between these points must be high enough to pick up under engine if required. These centres being fixed, a disc of wood is placed in outer end of stern tube, with a movable brass plate screwed to it. This plate has a centre hole, $\frac{1}{32}$ in. in diameter, which is adjusted in line with the other sight holes. Discs are now placed on forward end of A bracket, and inner end of stern tube, on after engine room bulkhead, and one is built up on the engine seating. All light is excluded, and centre holes are brought in line by so adjusting them that a beam of light from an electric or acetylene lamp can pass through them in succession. When all the centres are in line, circles are described from these centres for boring out the A bracket and the stern tube, and for fixing position of stern gland. This operation is carried out on a dark night, if circumstances allow.

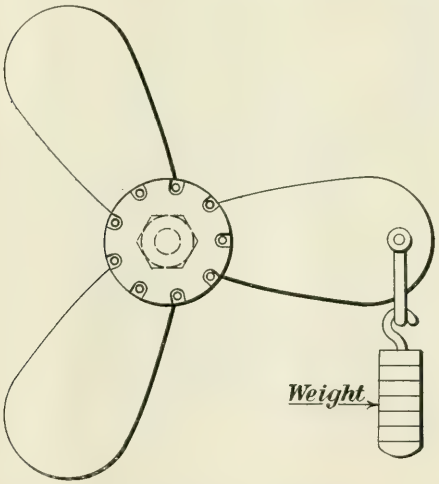
No. 4.—Setting a slide valve.

Two battens are taken, and from the end of one the following distances are marked:—Take distance from top flange to first port, and then a replica of the valve face showing ports.

This is mounted outside the valve and in juxtaposition with the valve face, by placing the end level with the top flange, and then the other batten is marked off with a replica of the valve, showing the lap. The engine is then placed on top dead centre, and the amount of lead is noticed, and the valve batten is placed in a corresponding position beside the valve face batten. The engine is turned through one-half revolution to the other dead centre, and the valve batten moved exactly the same amount that the valve rod has moved. The lead at the other end of valve should now be noted, and if the notations are not satisfactory, the valve must be set up on the rod or *vice versa* as may be required. By marking the centre of the travel of valve batten, this point should coincide with, or be a little above, the centre of the exhaust port. If the leads at both ends require to be increased or decreased to or from the designed amounts, this operation can only be accomplished by altering the angular advance of the eccentric.

No. 5.—Friction test of Shafting.

The friction test is always taken with a new job, both when the stern gland is packed and also when unpacked. It is a good plan to take this test each time a ship dry docks, and it should be noted on any reports as to condition of gland. The required weight when the ship is new is usually stamped in.

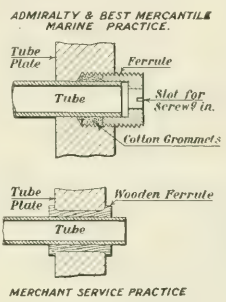


Diagrammatic Representation of Apparatus used for Friction Test for Shafting.

The procedure is as follows :—Remove the coupling bolts and place one blade of the propeller in a horizontal position. From this blade suspend weights until the shaft just turns. Tabulate this result with particulars and the radius of application of weight. An alternative method is to connect up with an eyebolt with blocks and tackle, and to interpose a Salter's balance or other instrument. Care must be taken in the record of test, however, to include weight of shackle and attachment.

No. 6.—Tubing or Re-tubing a Surface Condenser.

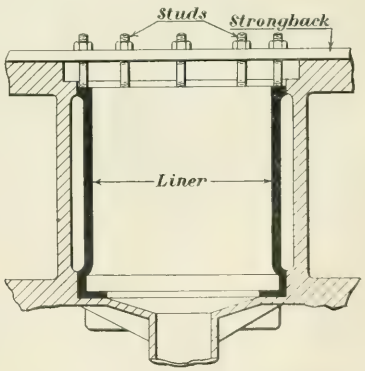
One of the tube plates is first placed in position, and the tubes are threaded through this and the diaphragm plate. The second tube plate is then placed in position, and the tubes are guided through their respective holes by means of a steel wire. They are then held individually in position at one end, while the specially prepared grommet, consisting of a few turns of tape with ends sewn over, soaked in boiled linseed oil, is slipped over the ends of the tubes, with the aid of a special coned mandril fitting into the ends of the tubes. Screwed ferrules, fitting over the tubes, are then screwed into the stuffing boxes in the tube plate, thus compressing the tape packing and making a watertight joint between the tube and tube plate. These ferrules are made with small internal projections or flanges, to prevent the tubes slipping out of place, and slots $\frac{1}{4}$ in. deep are cut across the face to enable one to screw up. In some vessels, however, a cheaper method is adopted in making the joint. Wooden ferrules fitting over the tubes are driven into plain holes in the plate, and when they come into contact with water, the ferrules swell and make the joint. The following defects however are liable to occur :—(*a*) If the screwed ferrules are screwed too tightly, the grommet may cause the tube to bulge in, thus diminishing the diameter of the bore and preventing the tube from expanding when it becomes heated ; (*b*) The ferrules having a fine thread are liable to be cross threaded when entering them (especially so when they are old), thus they do not make a good joint when they appear hard home ; (*c*) The bottom of stuffing boxes have sometimes given way when the ferrule has been screwed up too tightly. With modern appliances, however, the tube plates are now most accurately gauged, and this last seldom occurs.



Methods of Attaching ends of Condenser Tubes.

No. 8.—Removing Cylinder Liner.

Many cases have occurred in which a leak has been discovered between the jacket and cylinder, due in practically every case to the failure of the bottom joint. If a leak is suspected a water pressure test should be applied, when such a leak will at once be apparent.



Method of Removing Cylinder Liner.

The procedure is as follows :—First remove all the bolts from the bottom flange. Then the top ring is removed and the packing drawn. The quill for the indicator cock is next removed. Then screw the studs through clearing holes in strongback, into tapped holes on top of liner. Usually eight studs are used, although the more studs used the better, up to a certain limit. Then put nuts on the studs and screw up until liner is free. It should be noted that great care is necessary in removing the quill for the indicator cock, since this is fine threaded, and very liable to fracture.

CARGO BOAT MACHINERY.

Analysis of Machinery Installations as given in Lloyd's Register of Shipping.

In the article, "Cargo Boat Machinery," which we published in our February issue of 1922, we gave the rules formulated by the North-East Coast Institution of Engineers and Shipbuilders as given in their Guidance Specification, for finding the indicated horse-power of marine engines, also Lloyd's rule for the nominal horse-power of marine engines, and in our November issue, for the same year, we gave their rule for the nominal horse-power of marine natural draught boilers, together with rules for finding the size of the boilers for any given nominal horse-power. Table V has been compiled of machinery of six steamers constructed by various firms as given in Lloyd's Register.

To show the relation between the engines and boilers of any particular installation, it is necessary to find the nominal horse-power of the engines, also of the boilers, as the nominal horse-power given in Lloyd's Register is the nominal horse-power of the installation as a whole; this can easily be calculated by the rules given in our February and November issues previously referred to.

As the boiler is the dominating factor in any installation, it has been taken as the basis of power at the rate of 4·25 boiler nominal horse-power equal to the maximum indicated horse-power at sea, but not necessarily the actual indicated horse-power.

The size of the boilers and furnace work for each installation can be found by reference to the rules and tables in the November issue mentioned above. The total cylinder capacity is the capacity of each cylinder added together.

The installations have been placed in sequence of power irrespective of the year in which they were built, between 1900 and 1910.

The table is extended by the introduction of the several data relating to the engines and boilers so as to form a complete analysis of the installations from the particulars at our disposal.

Examine the nominal horse-power of the engines and boilers in reference No. III in Table VI below. Here the nominal horse-power of the boilers is less than the nominal horse-power of the engines; the large engine may be due to the desire of the owner to have a large engine, as some owners believes in a large slow-running engine so as to get a low consumption of coal, but this very often results in a great loss of speed in bad weather. Examine the furnace installation. The length of bar is partly an indication of the efficiency of the furnace, always remembering that the better the coal the shorter the fire bar should be; note its relative length to the length of the boiler.

The cylinder capacity being partly a measure of the engine, it is instructive to compare it with the grate surface of the boiler for each particular type of steamer.

The relation between the heating and grate surface of the boiler to the indicated horse-power of the engine are too well known to need comment.

The ratio of heating surface, it will be noted, is greater for the 160 lb. installations than the 180 lb. installations, and is also a measure of economy and efficiency for the particular type of steamers and of the kind of fuel used. This system of analysis can be of great service to chief engineers and superintendents of fleets of steamers; to the consulting engineer who thoroughly understands design, it can give the practice of each firm, and he can foresee what type of installation any particular firm of engineers will offer him when enquiring for new machinery. The method has been used with success by our correspondent for a period of over 30 years.

Table V.

Reference No.	I.	II.	III.	IV.	V.	VI.
Cylinders, H.I.L.	18½″–30″ 49″	24″–39½″–65″	25½″–40½″ 67″	25″–42″–68″	25½″–43″–70″	27″–45″–74″
Stroke and Working Press.	33″, 180 lb.	42″, 160 lb.	45″, 160 lb.	45″, 180 lb.	45″, 180 lb.	48″, 180 lb.
Nominal Horse-power	156	282	297	340	369	458
Boilers and Furnaces	1 SE, 4 f	2 SE, 6 f	2 SE, 6 f	3 SE, 9 f	3 SE, 9 f	3 SE, 12 f
Grate and Heat Surface sq. ft.	66, 2507	112, 4360	113, 4404	143, 5300	166, 5863	218, 7713
Draught	Natural	Natural	Natural	Natural	Natural	Natural

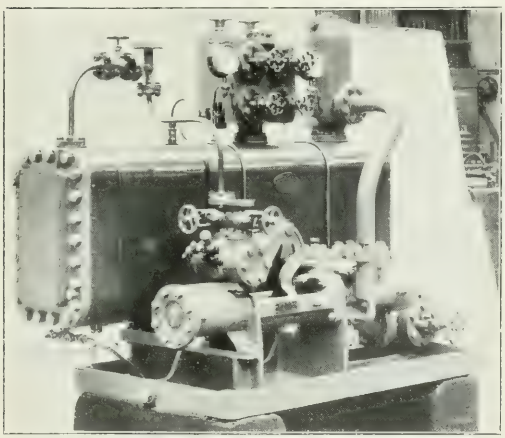
Table VI.—Analysis of Table V

Reference No.	I.	II.	III.	IV.	V.	VI.
Engine, Nominal Horse-power	141	274	301	318	346	390
Boiler, Nominal Horse-power	171	290	293	362	402	527
Ratio, B.N.H.P. ÷ E.N.H.P..	1·212	1·058	·975	1·138	1·16	1·35
Basis, I.H.P. = 4·25 B.N.HP.	727	1235	1250	1540	1710	2245
Boilers	16′ 0″ × 10′ 9″	15′ 0″ × 10′ 6″	15′ 0″ × 10′ 9″	14′ 0″ × 10′ 6″	14′ 6″ × 10′ 6″	16′ 0″ × 11′ 0″
Furnace and Bar	3′ 3¾″ × 5′ 0″	3′ 7″ × 5′ 3″	3′ 7″ × 5′ 3″	3′ 3¾″ × 4′ 9″	3′ 4¼″ × 5′ 6″	3′ 3¾″ × 5′ 6″
Total Cyl. Cap. ÷	54·7	122·2	139	143·3	151·5	180
Grate ÷ ÷ Cyl. Cap. ÷	1·208	·916	·8125	1·0	1·095	1·21
Heat ÷ ÷ I.H.P.	3·44	3·52	3 52	3·44	3·43	3·44
I.H.P. ÷ Grate ÷	11·0	11·02	11·05	10·77	10·3	10·3
I.H.P. ÷ Cy. Cap.	13·3	10·1	9·0	10·75	11·28	12·45
Ratio HS ÷ GS	37·9	38·9	38·9	37·0	35·3	35·35
Ratio L.P. Cyl.² ÷ H.P. Cyl.²	7·0	7·34	6·9	7·4	7·53	7·52
I.H.P. ÷ E.N.H.P.....	5·16	4·5	4·15	4·85	4·95	5·75

A NEW SYSTEM OF OIL FUEL BURNING.

A Brief Description of the Clyde Pressure Jet System fitted on the "Fezara" and "Famaka."

We illustrate on page 249 the Khedivial Mail liner *Fezara* which was built and engined by Alexander Stephen & Sons Ltd., Linthouse. She and her sister ship the *Famaka* are interesting in that their boilers are fitted with the Clyde pressure-jet system of oil fuel burning apparatus. This system, which is in use on several land installations was first adopted for marine work on board the *Famaka*. The method of working is to project the oil under pressure into the furnace in the form of a finely atomised conical spray by means of a special oil fuel pump which draws its supply from settling or storage tanks through a strainer of coarse mesh, where any foreign matter mixed with the oil is deposited and can, if necessary be removed periodically. The pump discharges the fuel through a heater, where its temperature and corresponding viscosity is brought to that necessary for complete atomisation in the burner. A discharge strainer of fine mesh is provided



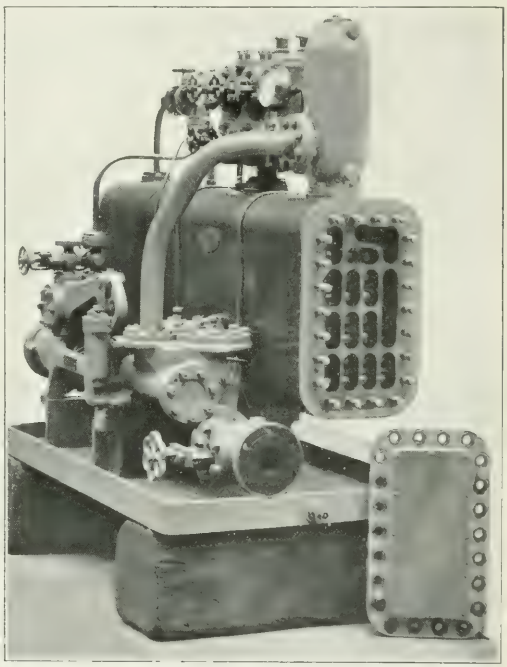
An External View of the Unit showing Pumps and Heater.

on the heater outlet to strain the oil finally before passing it to the burner. Both the suction and discharge strainers are of the duplex type, enabling one to be cleaned whilst the other is in operation.

Heating Plant.

The heating plants are generally fitted in duplicate, one working and one stand-by, but for small installations where the additional cost does not warrant the double plants, a combined plant can be supplied having a single heater with double pumps and the necessary shut-off valves to enable one pump to be overhauled while the other is working. Accessibility is one of the points in the design of the oil fuel unit to which particular attention has been paid, and it will be noticed in the plate showing the pumping and heating unit that the oil fuel pump has been brought out clear of the heater so as to give unrestricted head room for its attention and overhaul. The number of joints under oil pressure is reduced to a minimum, and where these exist they are such as to be accessible conveniently at all times.

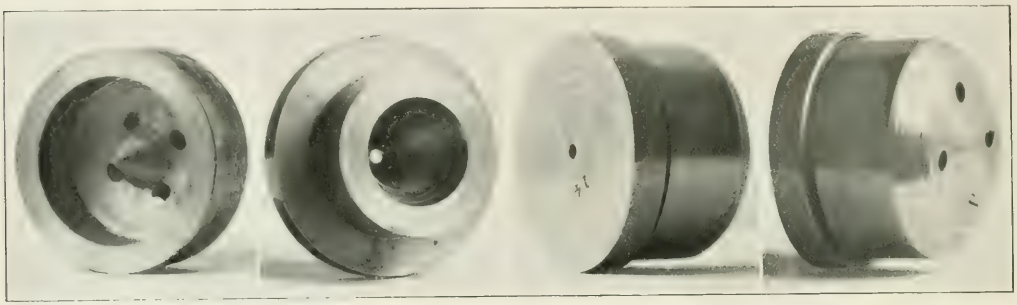
The Clyde patent oil fuel heater is of the coil type and possesses the advantage of this type in that the flow is in series through it, preventing silting up taking place, and also of the more efficient multitubular type, in that, should a tube give out, only a small portion of the heating surface becomes inoperative, thus rendering undue duplication of heaters unnecessary. The tube elements of the heater, illustrated on page 250, consist of a number of solid drawn steel tubes expanded into a tube plate at one end, leaving each tube free to expand individually. The tubes when assembled into the heater body are concentrically disposed within a series of circular passages interconnected at their ends in such a manner as to give to the



End View of the Unit showing Passages for Heater Tubes.

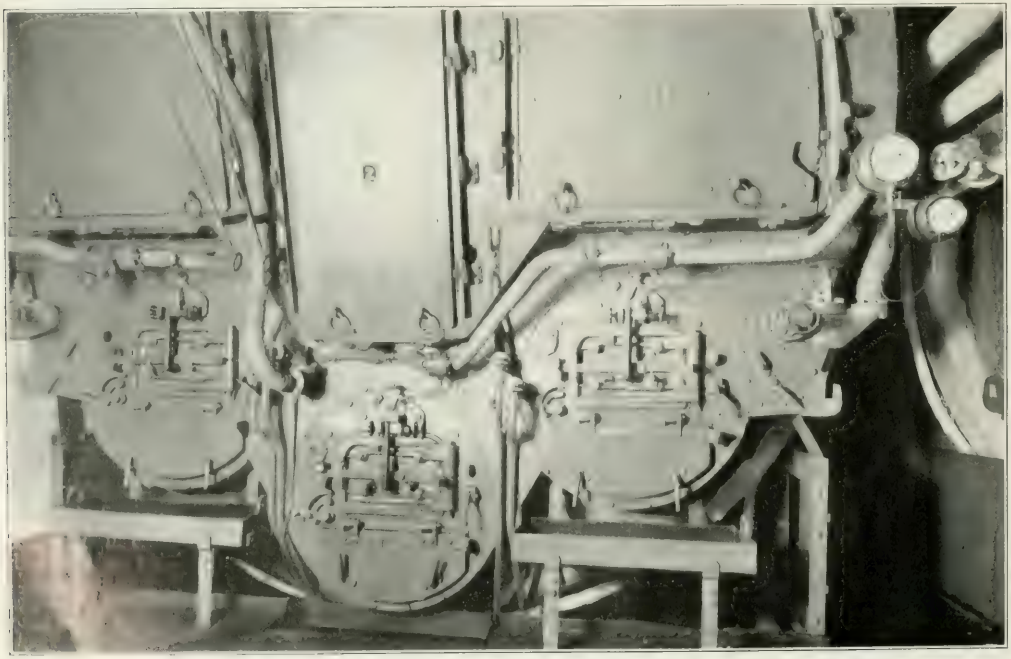
oil a sinuous path of travel through the entire heater body, at the same time providing for the examination, and, if necessary, cleaning of the oil space. Some of these heaters dealing with the heavy residue from stills have been in operation for long periods without requiring attention, and steaming out after use is unnecessary. All parts under oil-pressure throughout the system are tested to 400 lb. per square inch before leaving the maker's works. The sulphur, which most fuel oils contain in various proportions, when heated attacks the copper in brass, gradually rotting it; hence all the valves are designed to overcome this objection, being made of cast iron with steel fittings. The burners of the system are capable of atomising the oil completely at a pressure of 25 lb. per square inch and upwards, the range of output being varied through wide limits by increasing or decreasing the oil pressure, and the contour of the flame when working in conjunction with the air director remains constant throughout the pressure range.

When above or below the desired pressure limit, it is only necessary to substitute another size of nozzle, the diaphragms remaining the same throughout. The diaphragms and nozzles have been designed to present the minimum resistance in imparting to the projected oil the necessary centrifugal force to break it into a minutely atomised conical spray, in which condition it is in a position to receive the air required for combustion. A special feature of these burners is that the conical projection is completely shrouded, preventing damage to the point and ensuring a uniform spray with minimum replacement charges. The filtering of the oil being efficiently performed in the discharge strainer no further straining is required in the burners themselves. We illustrate the component parts of the atomiser elsewhere.



Views of the Component Parts of the Atomiser.

The Clyde patent air heating furnace front as fitted to marine boilers is illustrated on page 250. A special feature of this is that no brickwork in the furnaces or brick linings on the furnace front plate are required, a fact which reduces erection and maintenance cost. The regulated incoming air required for combustion is given a circulation over the furnace front plate, resulting in a higher initial flame temperature and earlier combustion, both of which ensure increased efficiency. The resultant flame is characterised by a dazzling whiteness at the centre, shading to orange, where it meets the furnace walls, being short and uniform without gaps and completely filling the furnace, thus eliminating the possibility of a non-conducting film of air being interposed between the periphery of the flame and the furnace walls and ensuring a high rate of heat transmission to the water. The design also ensures that no cold air can enter the furnace when working, thus prolonging the life of the boiler owing to the reduced temperature range. These furnace fronts are made of steel pressings throughout, ensuring immunity from distortion and cracking, and giving a minimum weight, making it easily handled when reverting



The Clyde System as applied to a Marine Boiler working under Forced Draught.

MARINE BOILER PLANT EFFICIENCY.

By DAVID BROWNLIE, B.Sc., A.M.I.M.E., A.I.Mech.E., Mem.Am.Soc.M.E.

A Criticism of Existing Methods of Boiler Plant Handling.

There is at least 10 to 12 per cent. of the coal bill waiting to be saved on the average marine boiler plant, chiefly by the adoption of more scientific methods of control, which in nearly every case can be carried out at a very small expenditure of time and trouble. I have been advocating these improved methods for years, but no steamship company in this country will pay any attention to the matter, and marine steam generation continues to be carried out on the crudest and most rule-of-thumb lines in spite of the severe competition represented by the Diesel engine. I am glad to notice, however, that if Great Britain continues to burn wastefully about 16,625,000 tons of her coal per annum in ocean-going and coasting steamers, matters are not quite so bad in the United States, where serious attention is at last being given to this question of marine steam generation. The principle of burning oil in marine boilers is, in my opinion, thoroughly unscientific from many points of view, and one of the most important of these—the fact that there is not sufficient oil in the world—has moved the United States Government to make a determined attempt to improve the efficiency of the existing steam plant in the American mercantile marine.

Oil Consumption.

The world to-day is consuming over 800,000,000 barrels of oil per annum, of which 66 per cent. is produced in the United States and about 20 per cent. in Mexico; British capitalists only control 5 per cent. of the oil of the world, whilst the whole British Empire only produces 2½ per cent. We have, in any case, all the oil we need at home in our own coal, but since we refuse to make the slightest attempt to extract it, and prefer to squander untold millions of money in Mesopotamia, it is not very good policy for British shipowners to extend the use of a fuel of which over 90 per cent. is controlled outside this country. But added to this is the grave fact that the petroleum supplies of the world, and especially the American wells, are failing, as the United States Geological Survey has been warning us for years, and it is for this reason the American Government is, quite rightly, so anxious to prevent the extension of the use of oil for marine steam generation by improving the steam plant. With this object the United States Navy has opened a school at the Philadelphia Navy Yard for the training of engineers and stokers, both in the naval and merchant services, in improved methods of steam generation. Also an extended investigation has been carried out at this station by the United States Shipping Board in conjunction with the Testing Staff of the Bureau of Mines, and some of the results obtained are given by Mr. C. L. Jefferson, the head of the Fuel Conservation Section of the United States Shipping Board, in a recent paper read before the American Society of Mechanical Engineers.

Log Analysis.

In the first place, it is stated that the above work included an analysis of the logs of approximately 250 American steamships, nearly all cargo vessels, and from these figures it is calculated that the average efficiency of the boiler plants, nearly all the ordinary Scotch marine type of boiler, is 60 to 65 per cent. with oil firing and 55 to 60 per cent. with coal firing, and the report goes on to say that the same state of affairs holds good for the average stationary plant of less than 1,000 boiler h.p. Now, in the first place, I am convinced that these figures are far too high, and the case for the necessity of improved methods of steam generation is even stronger than Mr. Jefferson has made out.

Scotch and Lancashire Boilers.

The Scotch marine boiler is very similar as regards the results obtained to the ordinary "Lancashire" boiler characteristic of British land practice in steam generation, and, as Mr. Jefferson states, there is little or no difference between the marine and the average industrial plant as far as the boiler is concerned. In the first place, with regard to the results now being obtained on land, during the past 15 years or so I have made a complete scientific investigation of the working of 400 different boiler plants in Great Britain in 41 separate industries burning 3,250,000 tons of coal per annum in 1,513 boilers, of which over 1,350 were of the "Lancashire" type. The true net working efficiency of all these boiler plants, boilers, economisers and superheaters was 58 per cent., whilst individual plants varied from 32·50 to 82·21 per cent. The figure of 58 per cent. was made up of approximately 53·25 per cent. due to the boiler, 4·25 per cent. to the economiser, and 0·50 per cent. to the superheater, these, of course, all being average figures and including many plants not fitted with economisers and superheaters. The point is that the average figure for the boilers only on land, representing many years' continuous work in testing, is only 53¼ per cent., and I think this can be taken as a very fair average, especially as far as my own experience goes, since I have examined, without testing, about 2,000 plants with a coal bill of 15,000,000 tons and found them all working on the same general lines. I have never actually tested a marine boiler at sea, but from the few investigations I have made of steamship boiler plants and a number of tests on the marine type of boiler installed on land, I should estimate the marine boiler is no more efficient than the "Lancashire" boiler. In view of the fact that nearly all marine boilers are being worked without superheaters, and economisers are never installed, whilst the average quality of the hand firing at sea is certainly no better than on land, I should say a fair figure for the coal-fired marine boiler plant is 52½ per cent., slightly less than the "Lancashire" boiler on land, which is a somewhat different matter from the 57½ per cent. (55 to 60 per cent.) of the American Shipping Board, representing 8½ per cent. in the coal bill.

Oil Firing.

As regards oil firing, I am not prepared to offer an opinion, except that many oil-fired plants are doing much worse than is commonly supposed. Whatever may be the true figure, it is agreed by all fuel experts that there is an enormous economy to be effected in the working of marine boiler plants. In this joint work of the United States Shipping Board and the Bureau of Mines, a large number of experiments have been carried out at the testing station with a Scotch marine boiler of the ordinary single-ended, three-furnace, separate combustion chamber type, having 2,777 sq. ft. of heating surface for coal firing and 3,022 sq. ft. when oil fired, the combustion chamber volume being 289·4 cubic ft. The boiler was also fitted with a "Foster" superheater of 774 sq. ft. heating surface, and an air heater of 1,220 sq. ft. for the hot air supply

for a "Howden" forced draught system with closed ashpits. Four distinct series of trials were carried out: (1) hand-fired with coal, (2) pulverised coal, (3) forced draught oil, and (4) induced draught oil, and in each case the work was continued until the most efficient results were obtained. Pulverised coal firing was, however, found to be of little importance because of the limited size of the furnaces of the Scotch boiler and the difficulty of obtaining the maximum heat, since the furnace

Results of Joint Tests Carried Out on a Boiler by the United States Shipping Board and the Bureau of Mines.

	Test Numbers.			
	1	2	3	4
Duration hours	12·00	10·00	8·08	6·09
Total coal fired lbs.	9,696	12,000	12,521	11,619
Coal per sq. ft. of grate area per hour	13·07	19·40	25·10	30·90
Analysis of coal :—				
(*a*) B.Th.U. per lb.	—	—	—	—
(*b*) Fixed carbon	70·93	69·87	70·00	69·44
(*c*) Volatile matter	18·84	19·07	19·52	19·14
(*d*) Ash	8·08	8·79	8·51	9·25
(*e*) Moisture	2·15	2·27	1·97	2·17
CO_2 in flue gases per cent.	11·2	12·2	12·3	11·8
Boiler pressure (gauge) .... lbs.	163	157	166	158
Superheat deg.	1	14	23	38
Flue gas temperatures :—				
(*a*) Leaving boiler, deg. Fahr.	437	503	537	606
(*b*) Leaving superheater, deg. Fahr.	372	447	472	519
(*c*) Leaving air heater, deg. Fahr.	277	336	361	396
Total water evaporated .... lbs.	100,560	124,670	122,328	116,988
Temperature of feed-water, deg. Fahr.	212	204	193	208
Actual evaporation per lb. coal ...	10·37	10·40	9·76	10·07
Efficiency of boiler without superheater, per cent.	75·2	76·7	72·6	74·6
Heat balance sheet :—				
Heat absorbed by the boiler	75·8	77·9	73·9	76·8
Loss due to moisture in coal	0·2	0·2	0·2	0·2
Loss due to burning hydrogen	3·4	3·5	3·5	3·6
Loss due to heat in dry gas	6·2	7·1	7·4	9·3
Loss due to CO_2 figure	0·4	0·9	1·3	0·1
Loss in ash and refuse	2·5	3·3	2·8	3·1
Loss not accounted for	11·5	7·1	10·9	6·9
Totals	100·00	100·0	100·00	100·00

is, of course, entirely water-cooled. It is obvious, however, that the question of marine pulverised fuel firing is a matter for future investigation, in view of the remarkable developments being made on land with the "Lopulco" system. With regard to ordinary coal firing, which is of most interest, the results obtained were remarkable, no less than 72½ to 76¾ per cent. efficiency with the boiler (and air heater) alone, and 74 to 78 per cent. with the superheater. The chief figures of four tests are given below, representing combustion ranges of from 13·07 to 30·90 lb. coal per sq. ft. of grate area per hour with the ordinary "Howden" forced draught hot-air system, as already described.

Efficiencies.

It seems to me to be hardly practical to expect marine boilers to be worked at 73·9 to 77·9 per cent. efficiency, these results being apparently exclusive of the superheater, but including the air heater. I have estimated that the average land boiler plant, boilers, economisers and superheaters, but without air heaters, now working at 58 per cent. efficiency, ought to work at 75 per cent., that is to say, Great Britain is losing, because of the lack of modern scientific methods of control, nearly 23 per cent. of the coal consumed in stationary and boiler plants, that is, over 20,000,000 tons in the present coal bill of 90,000,000 tons per annum. If we are going to assume, as a result of this work in America, that marine boilers with air heaters can work at 75·9 per cent. efficiency, then the saving to be obtained, even on the American figures of 57½ per cent. for the present performance, is actually 24¼ per cent. of the coal bill of the average steam-driven vessel, that is, 4,073,000 tons of coal per annum on the present consumption of 16,625,000 tons per annum in ocean-going and coasting steamers from British ports.

Certainly, in my opinion, a marine boiler plant with Scotch boilers, superheaters, and air heaters should work at 70 per cent. net working efficiency, which, on the American figures of 57½ per cent., would represent 17¾ per cent. saving in the coal bill, or 23·5 per cent. saving on my own figures of 53½ per cent., almost identical with the American figures. It may be stated also that as regards oil firing, the results obtained were equally remarkable, being, for example, 80·75 to 84·50 per cent. for the boilers only, with the forced draught system of firing.

Possibilities of Economy.

It is evident, therefore, that an enormous economy is possible in marine steam generation, and the chief reasons for this are very briefly as follows :—

In the first place, and most important of all, no attempt is made to keep any continuous record of the performance of the plant. Every marine boiler plant should be worked with a pressure type of boiler feed meter so as to record the evaporation, and in spite of the fact that such meters have been on the market for years, and one meter, the "Kennedy," for over 50 years, specifically introduced for marine work, there is hardly a single ship afloat to-day that keeps any record of the water evaporated in the boilers or the corresponding coal burnt. That is to say, the boiler plant is being worked in the dark as regards the performance, and this probably causes a loss of 5 per cent. in the coal bill alone. Then, again, there is hardly a single marine boiler furnace to-day being controlled with a combustion recorder, in spite of the fact that at least three different makes are on the market specially adapted for marine conditions.

It is impossible to obtain the best results in firing unless there is some means of knowing the efficiency continuously, and at least another 5 to 7½ per cent. in the coal bill is being lost on this account. Again, there is generally a considerable loss in efficiency due to general air leakage, and little use is made of superheating.

10 per cent. Saving Possible.

It can be stated therefore, without exaggeration, that at least 10 to 12 per cent. of the coal bill of the average marine boiler plant can be saved by adopting improved methods that are comparatively elementary, whilst if special care and attention is given the figure may easily be 20 per cent. for individual cases, and from the point of view of economy the boiler plant to-day is the most important part of any steamship.

NOTES ON THE ECONOMICAL PERFORMANCE OF TURBINE INSTALLATIONS IN MERCHANT SHIPS.*

The Location of Losses by the Measurement of Water Consumption.

In this paper the author gives a *résumé* of consumption trials which have been carried out in a few geared-turbine vessels under ordinary sea-going conditions, and deals with a few points affecting the efficient running of turbine installations under such sea-going conditions. The performance of cargo steamers is usually examined on the basis of coal consumed per day under voyage conditions. In the case of a poor performance the responsibility may lie with the main engines, with the auxiliary machinery, or with the boilers, but cannot be placed. It is the object of the paper to illustrate the usefulness and practicability of making measurements of water consumption which will in most cases enable losses to be located readily.

The results of three ships have been taken for the purposes of this paper, and for the sake of convenience the respective steamers are indicated by the letters "C," "S" and "T." In Table I, the general dimensions and particulars of the

TABLE I.—*General dimensions and particulars of Machinery for the Three Ships Considered.*

Steamer.	"C."	"S."	"T."
Type	2 Decks and shelter deck	3 Decks	2 Decks and shelter deck.
Gross tonnage	5,494.	9,773.	8,526.
Length B.P., breadth extreme, depth moulded.	425′ 0″ × 55′ 0″ × 37′ 3″ to shelter deck.	460′ 0″ × 62′ 6″ × 37′ 9″ to upper deck.	460′ 0″ × 62′ 6″ × 37′ 9″ to shelter deck.
Summer draught: displacement.	25′ 6¼″ 12,840 tons.	29′ 4″ 18,500 tons.	29′ 4″ 18,500 tons.
Shaft horse-power, designed	3,150 at 80 revolutions per minute	4,500 at 85 revolutions per minute	5,500 at 85 revolutions per minute.
Boilers	3 Single-ended.	4 Single-ended.	5 Single-ended.
Pressure ...	200 lb. per square inch.	190 lb. per square inch.	190 lb. per square inch.
Draught	Howden's forced draught.	Howden's forced draught.	Howden's forced draught.
Diameter: length	15′ 9″ × 11′ 9″.	17′ 6″ × 12′ 0″.	17′ 6″ × 12′ 0″.
Grate surface: heating surface	168 square feet; 7,863 square feet.	312 square feet; 13,900 square feet.	390 square feet; 17,375 square feet.
Number of furnaces	9	16	20
Superheaters	North Eastern Marine Smoke-tube type.	Smoke-tube type.	Smoke-tube type.
Superheat	200°	100	100°
Machinery, general description	One H.P. and one I.P. turbine in tandem, and one L.P. turbine; all geared by double-reduction gearing to one main shaft. All turbines reaction blading; H.P. fitted with "end-tightened blading"; air pump of the Dual type with steam jet augmentor; one centrifugal circulating pump with 2 single engines; pair of main feed pumps, direct acting, with control tank; cascade filter tank for auxiliary condensate only, one multiflow surface feed heater; two single direct-acting forced lubrication pumps; one fan 90″ diameter, with 2 single engines; two dynamos, one bilge and sanitary single direct-acting pump; one duplex ballast pump; one duplex general service pump; one single fresh-water pump; one single direct-acting auxiliary feed pump; one vacuum ash hoist with steam ejector; one 35-ton evaporator, and one Contraflo auxiliary condenser.	H.P. and I.P. turbines with individual pinions, geared to one primary wheel, and L.P. turbine geared to other primary wheel of double-reduction gearing to one main shaft; H.P. turbine, impulse reaction; I.P. and L.P. turbines, all reaction. Air pump of Dual type; one centrifugal circulating pump with two single engines; two single direct-acting main feed pumps; cascade feed filter tank for main and auxiliary condensates; one multiflow surface feed heater, two single direct-acting forced lubrication pumps, one centrifugal oil cooler pump, with single engine; two fans 84″ diameter, with single engine each; two dynamos with single engines; one duplex bilge pump; one duplex sanitary pump; one duplex ballast pump; one duplex general service pump; one duplex fresh-water pump; one single direct-acting auxiliary feed pump; one steam and vacuum ash hoist; two 25-ton evaporators, and one auxiliary condenser.	H.P. and I.P. turbines with individual pinions, geared to one primary wheel, and L.P. turbine geared to other primary wheel of double-reduction gearing to one main shaft; H.P. turbine impulse reaction; I.P. and L.P. turbines all reaction. Air pump, wet and dry, in conjunction with steam jet air ejector; one centrifugal circulating pump with two compound engines exhausting into main condenser; two single direct-acting main feed pumps; cascade feed filter tank for main and auxiliary condensate; one multiflow surface feed heater; two single direct-acting forced lubrication pumps; one centrifugal oil cooler pump with single engine; two fans 90″ diameter, with one compound engine each; two dynamos with compound engines; one single direct-acting bilge pump; one single direct-acting sanitary pump; one single direct-acting ballast pump; one single direct-acting general service pump; one single direct-acting fresh-water pump; one single direct-acting auxiliary feed pump; one ash ejector with single direct-acting pump; two evaporators and one Contraflo auxiliary condenser.

* Paper read by Mr. T. G. Potts before the North-East Coast Institution of Engineers and Shipbuilders.

machinery installation are given for each of the vessels taken. In steamer "C" the turbine engines are of the Parsons type, of the latest design, with reaction blading throughout, consisting of three turbines in series connected to double-reduction gearing, the high-pressure turbine being fitted with "end-tightening" blading. In this vessel measurements of water consumption were made by means of tanks, and also by water meters. In order that the water consumption of the main and auxiliary machinery might be separately measured and over the same period of time, the pipe connections were so arranged that the air pump discharged through a water meter to a temporary drain tank, and one of the main feed pumps drew the water from this drain tank and discharged to the measuring tank. All exhausts from the auxiliary engines were led to the auxiliary exhaust range and thence to the surface feed heater, and any surplus steam was led to the winch condenser. The drains from the feed heater, winch condenser, oil separator, cabin heating, &c., were led to the auxiliary cascade filter tank. A water meter was fitted in the drain pipe between the auxiliary filter tank and the feed pump control tank. The pipe and valve connections were so arranged that at all times both main and auxiliary water meters were in operation and the water from either the main or the auxiliary engines or from both could be pumped to the measuring tanks.

The contractors' trials of this vessel had to be carried out with the vessel in ballast condition, and it was necessary that the propeller be fully immersed in order that consistent torsion meter readings might be obtained. The vessel was specially trimmed by the stern to obtain this condition, giving an immersion of 10 in. at the propeller tip. The Contractors' trials were as follows:—

Trial No. 1.—Sunderland to Antwerp, July 21 and 22, 1921.

Trial No. 2.—Antwerp to Tyne, August 19 and 20, 1921.

Trial No. 3.—Progressive trial off Hartley, August 23, 1921.

Owing to the lengthy period the vessel was lying in port between the date of launching and the trials, and also between the first and the second trials, it is interesting to note in Fig. 1 the effect of the state of the hull upon the revolutions when the main machinery was developing the same power, and this difference is further shown by the results between the second and third trials due to the dry docking and painting after the second trial.

TABLE II.—*Readings and Data for Water Consumption Trials Nos. 1 and 2.* (*See page* 256.)

Draught forward	10′ 9″	Propeller diameter	18′ 6″
Draught aft	20′ 6″	Propeller pitch	16′ 9″
Draught, mean	15′ 7½″	Propeller, expanded area	100 sq. ft.
Displacement	7,465 tons.		

	Contractors' Trial No. 1—July 21 and 22, 1921. Sunderland to Antwerp.				Contractors' Trial No. 2—August 19, 1921. Antwerp to River Tyne.	
Date	21–7–21	22–7–21	22–7–21	22–7–21	19–8–21	19–8–21
Pressure at boilers	198	200	193	200	195	182
Pressure and superheat, auxiliary steam.	198 86°	200 111°	193 108°	200 84°	195 83°	182 86°
Pressure and superheat, main steam.	192 162	197 174°	186 184°	183 200°	173 197°	173 202°
Pressure and superheat, H.P. first expansion	55 198°	81 202°	110 204°	150 204°	150 206°	165 207°
Pressure, I.P. first expansion	7·27	15·8	27	42	40·3	45·5
Pressure, L.P. first expansion	17·95″	12·45″	5·5″	1·4 lb.	1·13 lb.	2·4 lb.
Vacuum at L.P. exhaust	28·9″	28·74″	28·46″	27·87″	27·95″	28·26″
Barometer	30·0″	29·95″	29·88″	29·84″	30·02″	30·1″
Pressure auxiliary steam (reduced)	108	108	109	109	149	154
Pressure auxiliary exhaust	6	5·5	8	10	5	9·5
Auxiliary exhaust led to	Winch condenser	Winch condenser	Winch condenser and feed heater.	Winch condenser and feed heater.	Winch condenser and feed heater.	Winch condenser. and feed heater
Revolutions per minute	54·19	61·66	68·58	76·2	73·18	75·18
Shaft horse-power	988	1,545	2,132	2,933	2,885	3,165
Temperature of sea	58°	58°	62°	60	65°	61·5°
Temperature of circulating discharge.	72°	74°	76°	78	86°	84·5°
Temperature of wet air pump suction.	76·5°	80°	79°	84°	92°	91°
Temperature to feed to boilers	103°	103°	158°	211°	216°	204°
	lb./s.h.p.	lb./s.h.p.	lb./s.h.p.	lb./s.h.p.	lb./s.h.p.	lb./s.h.p.
Water, lb. per hour, turbines	11,920 12·07	16,180 10·47	20,790 9·75	26,750 9·12	26,565 9·207	28,660 9·055
Water, lb. per hour, auxiliaries	3,170 3·21	3,430 2·22	3,770 1·768	4,530 1·544	4,485 1·555	4,730 1·494
Water, lb. per hour, turbines + auxiliaries.	15,090 15·28	19,610 12·69	24,560 11·518	31,280 10·664	31,050 10·762	33,390 10·549

The results of the water-consumption trials Nos. 1 and 2 are shown on Fig. 2 plotted on the basis of shaft horse-power. The actual readings and data of these trials are given in Table II. The results of the progressive trial No. 3 are shown in Fig. 3,

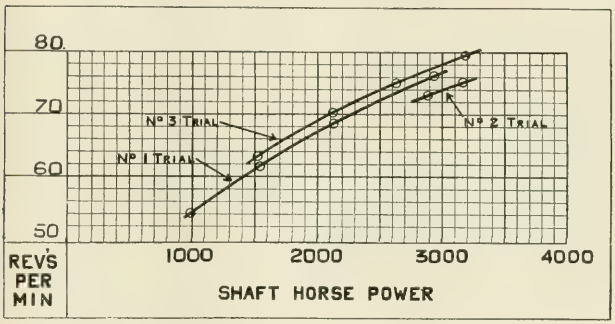


Fig. 1.—Showing the Effect of State of Hull upon R.P.M. with same S.H.P., Steamer C.

plotted to a base of speed and actual readings in Table III. The effective horse-power shown on this diagram is the effective horse-power as deduced from model experiments (naked hull).

Since the vessel first went into service over twelve months ago, the engineering staff of the vessel have taken full records of the performance of the propelling machinery on

and water consumption at the beginning of one of the watches. The trial continued under these conditions until the coal was used up. The data of the results of these trials are given in Table V. In Table VII the coal consumption results of eight outward and home voyages are given (home voyages loaded) during the same period, as taken from the engineers' log.

To obtain consistent readings it is necessary that the steaming conditions should be kept as steady as possible. Unfortunately, with coal-fired boilers, and the varying conditions obtaining at sea, it is not reasonable to expect a steady pressure of steam for 24 hours together. To test the efficiency of the plant during a voyage, it was arranged to run a short water-consumption trial (say of two hours) once every week, and to fix the following conditions :—

Initial pressure at the high-pressure turbine to be maintained constant throughout this period; no make-up feed to be used, and ash-hoist engine shut off; the temperature of steam at the high-pressure turbine to be maintained as constant as practicable.

During these tests, the consumption of the main and and auxiliary machinery was measured separately. A specimen of the data obtained under these conditions is given in Table IV, and illustrates the value of carrying out systematic tests.

Taking the results obtained on November 5, 1921, and comparing them with those obtained on the contractors' trials, it will be observed that at the same pressure and superheat there

TABLE III.—*Contractors' Trial No.* 3. (*See page* 257.)

Progressive Trial on Hartley measured course, August 23, 1921.

Draught, forward.... 10′ 9″ | Weather: Wind, fresh N.N.E.; Sea, slight swell. | Propeller diameter .... 18′ 6″
Draught, aft .... 20′ 6″ | Temperature of sea .. 58° Fahr. | Propeller pitch .... .. 16′ 9″
Draught, mean .... 15′ 7½″ | Displacement.... .... 7.465 tons. | Propeller, expanded area 100 square feet.

Direction of run	North.	South.	Mean.	North.	South.	Mean.	North.	South.	Mean.	North.	South.	Mean.
Speed in knots	10·1237	11·3421	10·7329	11·5606	12·0805	11·8205	12·7298	12·2866	12·5082	12·0805	13·9211	13·0008
Revolutions per minute	63·16	63·4	63·28	70·0	70·75	70·375	75·75	74·4	75·075	79·4	79·55	79·475
Slip per cent., apparent	—	—	—2·6		—	—1·6	—	—	—0·8	—	—	1·03
Shaft horse-power	1,531	1,523	1,527	2,119	2,133	2,126	2,750	2,510	2,630	3,191	3,179	3,185
Pressure at boilers	175	175	175	160	160	160	155	155	155	200	200	200
Pressure and superheat, main steam	171 175°	165 175°	168 175°	149 187°	154 187°	151·5 187°	154 194°	142 97°	148 195·5°	194 202°	187 202°	190·5 202°
Pressure and superheat, H.P. first expansion.	80 206°	80 206°	80 206°	110 203°	110 203°	110 203°	144 203°	132 201°	138 202°	165 209°	165 209°	165 209°
Pressure, L.P. first expansion ..	17·6	17·6	17·6	28·0	28·0	28·0	38·5	34·8	36·65	45·7	45·8	45·75
Pressure, L.P. first expansion	9·0″	9·0″	9·0″	4·0″	4·0″	4·0″	0·5 lb.	0·8″	0·05 lb.	2·0 lb.	2·1 lb.	2·05 lb.
Vacuum at L.P. exhaust	28·84″	28·76″	28·80″	28·53″	28·58″	28·55″	28·31″	28·39″	28·35″	28·09″	28·07″	28·08″
Barometer	—	—	30·04″	—	—	30·04″	—	—	30·04″	—	—	30·03″
Auxiliary exhaust pressure	15·0 lb.	14·0 lb.	14·5 lb.	10·0 lb.	9·0 lb.	9·5 lb.	5·5 lb.	5·5 lb.	5·5 lb.	4·0 lb.	5·0 lb.	4·5 lb.
Auxiliary exhaust led to	Feed he win	ater and ch conde	excess to nser.	Feed he win	ater and ch conde	excess to nser.	Feed he win	ater and ch conde	excess to nser.	Feed he win	ater and ch conde	excess to nser.

the several voyages. During this period, the vessel had steamed a distance of about 50,000 miles. During the outward passage in ballast, coal and water-measuring trials were carried out by the engineers of the vessel to determine boiler efficiency, the procedure adopted being to ship about 10 days' supply of coal into the cross-bunker, and after the vessel was clear of the port and the steaming conditions were quite steady, a start was made to measure the coal

is an increase of steam consumption of the main condensate of 1,530 lb. per hour—equivalent to 6¼ per cent. of the main condensate. To locate this increase, a trial was run between the Tyne and Leith on November 17, which confirmed the chief engineer's figures of an increase of over 6 per cent.

It was found that the valve on the drain connection from the high-pressure turbine first expansion belt to main condenser was leaking. On opening up, it was found that the valve was

badly eroded. This valve and seat were renewed, and on a subsequent test, the consumption for the main condensate over and above that obtained on the contractors' trials was only 390 lb. per hour, or 1½ per cent.

Another source by which the steam consumption is increased is in the case where there is air leakage to the main condenser resulting in a drop in vacuum, which necessitates driving the air, and particularly the circulating pumps, at such a speed that the increase in the consumption of the auxiliary machinery is out of all proportion. As is probably well known, the circulating pump has the largest duty of all the auxiliary pumps, and in most cases is fitted with a single-cylinder engine exhausting against a back pressure of 10 lb., which can hardly be termed an economical engine.

In the case of "C" ship, a test was made on the effect of increasing the revolutions of the circulating pump by about 50 per cent., but it was found that whilst a higher vacuum was obtained, the increased consumption of the circulating engine practically neutralised the gain in turbine efficiency due to the increase in vacuum. This points to the desirability of a more economical circulating engine, such as a compound engine, to gain the benefits of a high vacuum.

In this ship it will be noted from the data shown in the diagram that the steam consumption of the turbines only, was at the beginning of the vessel's life 9·3 lb. per shaft horse-power at 2,700 h.p., the steam pressure at turbines 139 lb. gauge, 204 deg. of superheat, the vacuum 28·21 in., and that the steam for the auxiliaries was 1·55 lb. per shaft horse-power.

After the vessel had been in service for over twelve months, a water-measurement trial was carried out, and the water consumption of the turbines only, was 9·35 lb. per shaft horse-power with steam pressure at turbines at 138 lb. gauge, 200 deg.

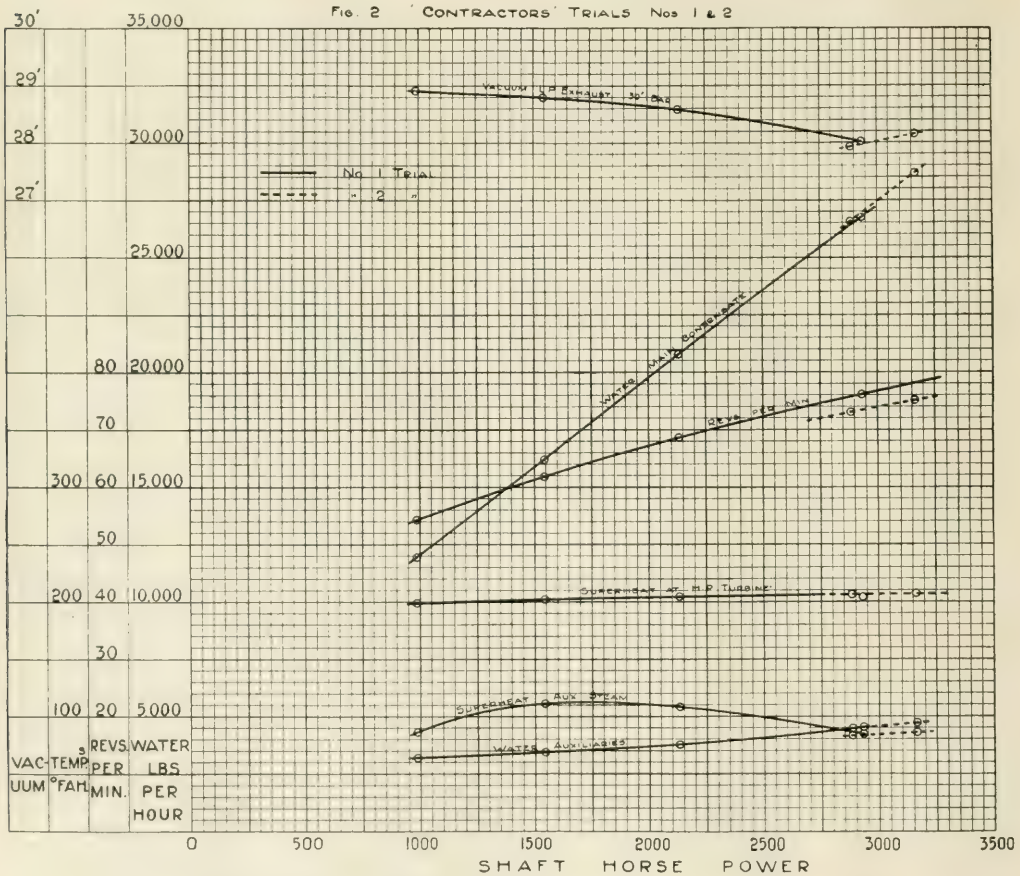


Fig. 2.—Results of Water Consumption Trials Nos. 1 and 2 to a Base of S.H.P., Steamer C.

Reverting to the results of coal consumption of voyages, given in Table VIII, as some engineers attach considerable importance to the value of the coal coefficient in making comparisons with other ships, the value of the coal coefficient has been added. It will be interesting to note that whilst the coal per shaft

horse-power is practically identical at both light and load conditions, there is a considerable difference in the actual coal coefficient between the light and the full load condition when the machinery is developing practically the same shaft horse-power. This would appear to serve as an illustration that considerable care requires to be exercised in making comparisons on the basis of the coal coefficient. The condition must, of course, be correspondingly similar.

It may be of interest at this stage to refer to the question generally of auxiliary machinery in merchant ships, for whilst

TABLE IV.—*Trials showing Effect of Leaky Drain Valve.*

Date of trial	5-11-1921	17-11-1921
High-pressure first expansion gauge pressure	135	140
High-pressure first expansion superheat Fahr.	199 deg.	190 deg.
Low-pressure exhaust vacuum at 30 in. barometer	28·6 in.	28·28 in.
Water, lb. per hour—		
Main condensate	25,980	26,830
Main condensate obtained on Contractors' trials	24,450	25,155
Difference between voyage and trials	1,530	1,675
Percentage increase due to leaky drain valve	6·26	6·66

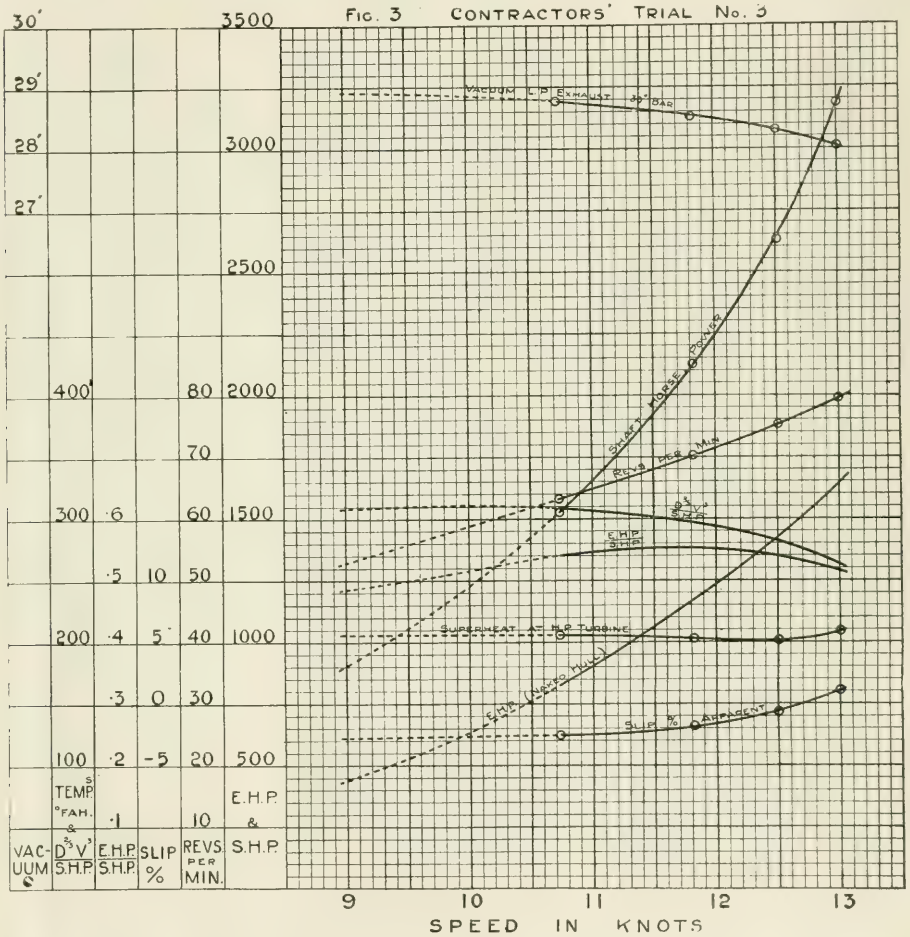


Fig. 3.—Graphical Representation of Results of Trial No. 3.

economy in steam consumption has been increasingly effected in the main turbines, the same can hardly be said of the auxiliary engines which form an important part of the propelling machinery in merchant ships. Reliability and economy in working are undoubtedly the two most important factors of auxiliary engines; piston engines are favoured by many engineers. Some auxiliary engines of this type are expensive in steam consumption, and although manufacturers have done much to improve them, there appears room for still further improvement in this respect.

A closer attention might also be given by the designer to a more suitable arrangement in the ship and to suitable pipe leads. In many cases it would appear that further economies might be effected by the sea-going engineer in a more economical running of the auxiliary plant. It is true that the exhaust

from auxiliary engines can be utilised to heat the feed water to an economical temperature, but in many cases a much greater quantity of exhaust steam is available than is necessary to heat the feed water, to, say, 220 deg., the surplus being led to the main or winch condenser. The alternative is to lead the surplus to the low-pressure turbines. It is common practice in naval work to utilise the auxiliary exhaust in the low-pressure turbines, when about 50 per cent. of the available energy is recovered.

This arrangement has been adopted in many merchant ships, instead of a separate pump as formerly used. On the next voyage made by this vessel, when steaming at full power, the auxiliary consumption was reduced by nearly 1,000 lb. as compared with the total quantity of auxiliaries when running at only two-thirds power. The percentage of auxiliary to main engines, plus drains, was about 20 per cent. at full power as against 33 per cent. at two-thirds power. There was still, however, more than sufficient to heat the feed water as before to 214° Fahr.

The ideal condition for merchant ships appears to be that

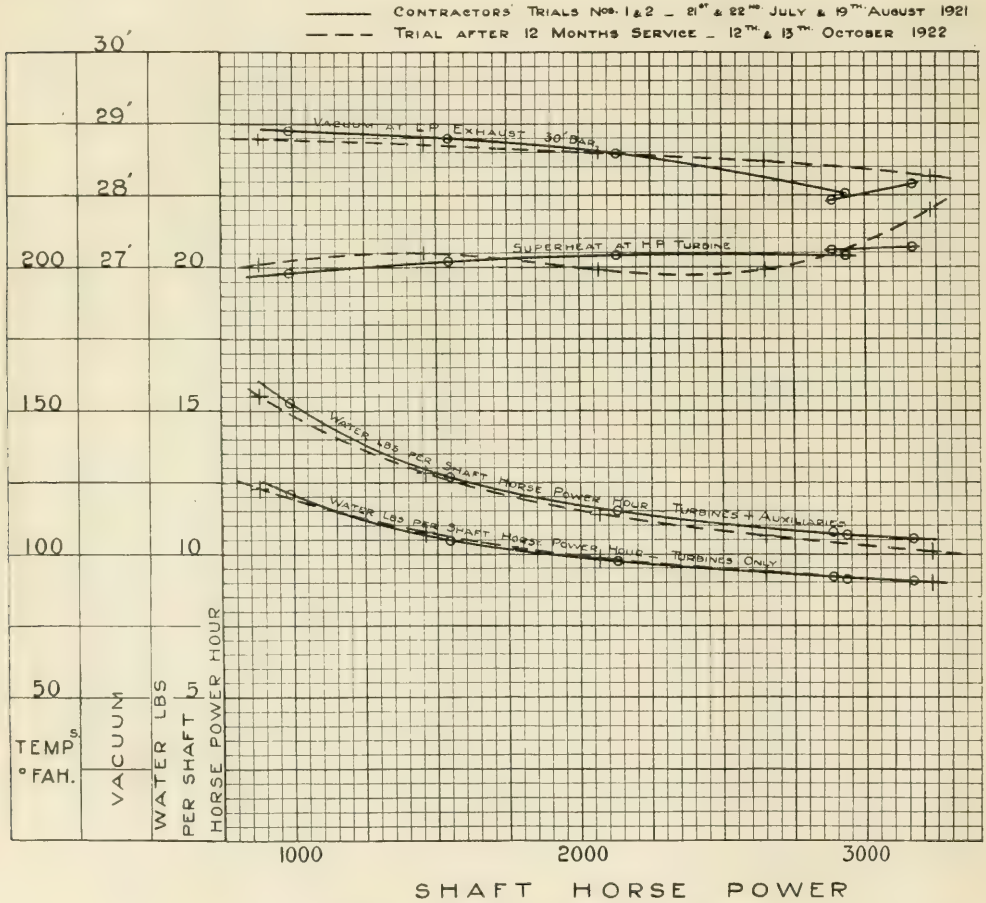


Fig. 4.—Ship C Trial Results after 12 months on service.

and in some instances has been found to prove disadvantageous where oil has been carried over with the steam from internal lubrication of the auxiliaries, which tends to leave a deposit on the blades of the turbine and the condenser tubes.

In column "B" of Table VII, it will be observed that when running under the same conditions as "A" there was sufficient auxiliary exhaust steam led to the feed heater to give 214 deg. feed temperature, leaving a surplus of 5,000 lb. per hour still passing to auxiliary condenser. After this voyage, the auxiliary engines were overhauled, and modifications were made so that the oil coolers were circulated by the main circulating pump where the consumption of auxiliaries is reduced to such an amount that all the exhaust can be utilised in feed heating. In the vessel "C" just previously described, the whole of the auxiliaries are now utilised in feed heating.

Referring to the boiler efficiency trials given in Table V, the main condensate includes make-up feed which was taken from the reserve tanks and led to the condenser, so as to maintain a constant boiler level. This question of lost water has an important effect on the general economy, especially if the evaporator has to be used to restore it.

As illustrating the effect on the coal bill of two vessels, viz., "S" and "T," which came under the author's notice, it may

be mentioned that in the case of "S" running at two-thirds power, the boiler feed as measured was 44,000 lb. per hour; the steam for auxiliary engines was 11,150 lb. per hour, leaving 32,950 per hour for main engines, plus all drains from cabin heating, galley, &c., which were led to the filter tank. The percentage of auxiliary consumption to main engines, &c., was 33 per cent.

To show the effect of fitting different types of auxiliary engines, the vessel "T" is given, being a sister ship to "S," but of increased power. The circulating pump engine was of the compound type exhausting direct into the main condenser, also supplying circulating water to the oil coolers. The forced-draught fan engines were also of the compound type exhausting into the auxiliary range. During one of the voyages a trial of eight hours' duration was carried out, and the results are given in Table VI. It is interesting to observe that in this instance the whole of the steam from the auxiliary exhaust range was utilised in the feed heater, the temperature of the feed to the boilers being only 194 deg. the water consumption for turbines and auxiliaries being 10·95 lb. per shaft horse-power.

The effect on the feed temperature and auxiliary exhaust pressure when the ash ejector was in use is also worth noting, the feed temperature rising from 202 to 215 deg., and the pressure rising from 3 to 11 lb.

During the last few years it has been the author's privilege to have had the opportunity of investigating the performances of a number of geared-turbine steamers, and in some cases where high consumptions of fuel have been recorded, it has been regrettable to note the poor efforts put forward to locate the reasons for them. Quality of fuel, poor class of fireman, and other similar reasons have been advanced, and whilst to a certain extent these have been contributory causes, there are many cases where more careful investigation—such as water-measuring the main engines and auxiliaries separately—would have led to a correct location of the main causes of the apparent high consumptions.

It is appreciated that the installation of measuring tanks may involve expense which some shipowners may not be prepared to face, but the fitting of meters for measuring the water can be carried out at a nominal cost, and the advantages of such measurements are of great value to the shipowner. It is true

TABLE VI.—*Steamer "T." Auxiliary Trials.*

Duration of trial	8 hours.
Pressure at boilers	190.
Pressure and superheat at high-pressure nozzle box	163 118°
Pressure, high-pressure reaction, first expansion	87.
Vacuum at 30 in. barometer	28·9 in.
Revolutions per minute	86·9.
Shaft horse-power	5,750.
Auxiliary exhaust pressure	3·5.
Auxiliary exhaust steam led to	Feed heater, excepting that circulating engine exhausts into main condenser.
Air pumps, D.S. per minute	22.
Circulating pump, revolutions per minute	154.
Forced lubrication pumps (2), D.S. per minute	21.
Fan engines (2), revolutions per minute	340.
Dynamo engine	One.
Main feed pump, D.S. per minute	7·35.
Ballast pump, D.S. per minute	Not in use.
General service pump, D.S. per minute	Not in use.
Oil cooler pump, D.S. per minute	Not in use.*
Bilge pump, D.S. per minute	Not in use.
Sanitary pump, D.S. per minute	22.
Ash ejector pump, D.S. per minute	As required.
Steering engine, D.S. per minute	As required.
Temperature circulating inlet	48 deg.
Temperature circulating discharge	68 deg.
Temperature feed discharge to boilers	194 deg.
Make-up feed from tank	Water level in boilers brought to same at finish as at start of trial.
Water, lb. per hour—	
Turbines, plus auxiliaries	63,000.
Make-up from feed tank	850.
Boiler feed	63,850.
Water, lb. per shaft horse-power per hour—	
Turbines, plus auxiliaries	10·95.
Make-up feed from tank	0·15.
Boiler feed	11·10.

* Oil cooler circulated from main circulating pump.

NOTE.—The following data shows the effects on auxiliary exhaust pressure and feed-water temperature due to ash ejector being in use.

	Off.	On.
Ash ejector		
High-pressure first expansion pressure	87	87
Auxiliary exhaust pressure	3	11
Temperature—		
Feed heater inlet	93°	99°
Feed heater outlet	202°	216°

TABLE V.—*Boiler Efficiency Trials on Voyages.*

Voyage No.	2	3	5	6	7	8
Duration, hours	187·783	176	261·25	235·4	197·5	212·2
Boiler pressure, lb. per square inch	188	176	190	190	190	190
Temperature, superheated steam, main	533°	566°	553°	556°	549°	551°
Temperature, superheated steam, auxiliary	450°	440°	425°	442°	468°	435°
Feed temperature	219°	220°	218°	219°	205°	200°
Water, lb. per hour— Main condensate (including make-up feed)	26,580	23,830	25,530	25,310	25,610	26,930
Auxiliary condensate	5,170	5,870	4,150	5,310	3,580	3,500
Boiler feed	31,750	29,700	29,680	30,620	29,190	30,430
Coal, lb. per hour	3,645	4,010	3,475	3,275	3,262	3,330
Kind of coal	Scotch.	Scotch.	Scotch.	Scotch.	Scotch.	Scotch.
Calorific value, B.Th.U.'s (gross)	14,338	14,263	14,200	14,120	14,038	13,587
Evaporation at feed temperature	8·71	7·41	8·54	9·35	8·95	9·14
Evaporation from and at 212°	9·93	8·4	9·66	10·58	10·28	10·52
Boiler efficiency, per cent.	67·2	57·1	66·0	72·7	71·0	75·1
Ashes, percentage of coal	15·2	18·1	16·4	12·8	12·9	13·5

TABLE VII.—*Steamer "S." Trial Data.*

	A.	B.	C.
Pressure at boilers .... lb.	180	181	180
Pressure and superheat at H.P. nozzle box	102 lb. 166	102 lb. 166	150 lb. 102°
Pressure, H.P. reaction first expansion lb.	60	60	102
Vacuum at 30" barometer	28·1"	28·1"	27·3"
Revolutions per minute	70·76	70·96	80·75
Shaft horse-power	2,960	2,960	4,491
Auxiliary exhaust pressure .... lb.	6·5	6·0	2·0
Auxiliary exhaust steam led	All to auxiliary condenser, auxiliary drains, galley and steam heating drains to filter tank.	To feed heater and surplus to auxiliary condenser, auxiliary drains, feed heater drain, galley and steam heating drains to filter tank.	All to auxiliary condenser, auxiliary drains, galley and steam heating drains to filter tank.
Air pump, D.S. per minute	26·9	27·5	32·0
Circulating pump, revolutions per minute	105·6	104	165
Forced lubrication pumps (2), D.S. per min.	17	22	14
Fan engines (2), revolutions per minute	242	230	280
Dynamo engine	One.	One.	One.
Main feed pump, D.S. per minute	6·15	6·26	9·04
Ballast pump, D.S. per minute	30·37 circulating aux. condenser.	31·0 circulating aux. condenser.	32·0 circulating aux. condenser.
General service pump, D.S. per minute	20·6	22·4	Not in use.
Oil cooler pump, D.S. per minute	172·4	187·5	Not in use.
Bilge pump, D.S. per minute	—	—	12
Sanitary pump, D.S. per minute	105	98·3	48
Steering engine	As required.	As required.	As required.
Temperatures—			
Circulating inlet and discharge	51·5° 68°	51·5° 68°	74° 91°
Feed discharge to boilers	84°	214°	106°
Make-up feed	Nil.	Nil.	Nil.
Water, lb. per hour—			
Main engines, plus all drains to filter tank	32,950	39,300	51,039
Auxiliaries to filter tank	—		—
Auxiliaries to auxiliary condenser	11,150	5,000	10,211
Boiler feed	44,100	44,300	61,250
Water, lb. per S.H.P. per hour—			
Main engines, plus all drains to filter tank	11·13	13·277	11·36
Auxiliaries to filter tank	—		—
Auxiliaries to auxiliary condenser	3·77	1·69	2·28
Boiler feed	14·90	14·967	13·64
Percentage of auxiliaries to "main engines, plus all drains to filter tank."	33·8		20

that water meters are looked upon with a certain amount of suspicion as regards their reliability, but when properly looked after and precautions are taken at intervals against errors, by calibrating the meters, they are found to be very reliable. Where the measurements were carried out in vessel "C." referred to in the paper, the engineers took a particular interest in this work, and it enabled them to place their fingers on any defects in the system which they were unable to locate by casual observation.

The author thanks those shipowners and superintendent engineers who have, from time to time, granted facilities for the purpose of these investigations.

TABLE VIII.—*Vessel "C". Results of Eight Outward and Home Voyages. (See previous page).*

Voyage No.	1	Outward 2	Homeward. 1	Homeward. 2	Outward 3	Homeward. 3	Outward 4	Homeward 4	Outward 5	Homeward 5	Outward 6	Homeward 6	Outward 7	Homeward 7	Outward 8	Homeward 8
Mean draught	14′ 10¼″	15′ 9″	24′ 5¾″	24′ 0¼″	15′ 9¾″	20′ 6¾″	15′ 11½″	24′ 5″	16′ 3½″	25′ 0¼″	16′ 3½″	23′ 6″	16′ 6″	25′ 1″	16′ 3¾″	25′ 0¾″
Mean displacement, tons	7,065	7,537	12,268	12,290	7,567	10,140	7,648	12,240	7,836	12,570	7,828	11,730	7,945	12,600	7,853	12,585
Distance, miles	3,196	2,901	3,220	2,908	2,903	3,107	3,162	3,230	3,201	3,126	3,151	3,124	2,757	2,907	2,861	2,589
Steaming time, days	11·25	11·106	11·66	10·873	11·518	11·173	13·187	12·672	12·363	11·129	11·85	11·39	9·79	10·546	10·292	9·64
Revolutions per minute	74·0	73·9	73·35	73·25	74·7	73·16	71·17	70·4	74·63	73·94	71·94	71·6	73·5	72·63	74·6	74·26
Shaft horse-power	2,490	2,641	2,550	2,608	2,351	2,512	2,623	2,626	2,709	2,825	2,739	2,764	2,701	2,715	2,763	2,834
Speed in knots	11·83	10·88	11·5	11·15	10·5	11·58	10·0	10·57	10·79	11·7	11·08	11·43	11·72	11·49	11·58	11·19
Slip per cent. (apparent)	3·2	10·9	5·1	7·9	14·9	4·2	14·9	9·0	12·5	4·2	6·8	3·4	3·4	4·4	6·0	8·8
Coal per day, tons	38·0	39·6	37·0	39·7	40·4	39·7	39·9	40·8	37·1	37·7	34·7	35·7	35·0	36·3	36·5	35·8
Coal, kind used	Durham.	Scotch.	Canadian and Durham	Scotch.	Scotch. Unscreened, very dirty.	Scotch. Unscreened, poor and fair.	Scotch. Poor and dirty.	Scotch. Bad.	Scotch. Good steaming.	Scotch. Good to fair.	Scotch. Fair and clean.	Durham Fair steaming, small.	Scotch. Good	Scotch. Good steaming.	Scotch. Good steaming.	Scotch. Good steaming.
Coal, per s.h.p. hour lb.	1·425	1·399	1·353	1·389	1·602	1·475	1·42	1·45	1·278	1·245	1·182	1·205	1·209	1·248	1·231	1·179
Percentage of ash	—	12	—	9·2	18	16	23	21	15	16·4	12·8	14·6	12·9	12·25	13·5	13·0
$\frac{D^{\frac{2}{3}} \times V^3}{S.H.P.}$	245	187·5	217·5	277	189·5	290	148	239	183	306·5	196	279	237·5	302·5	222	267
Coal per day	16,050	12,500	21,900	18,610	11,050	18,350	9,720	15,400	13,350	22,960	15,450	21,600	18,340	22,620	16,820	21,150

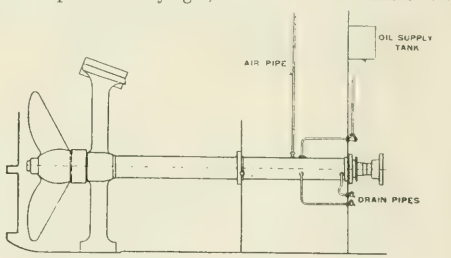
THE LUBRICATION OF PROPELLER SHAFTS OF HOPPER BARGES.

Description of an Appliance for Excluding Sand.

Some interesting particulars regarding the method employed by Mr. Daniel Fife (Chief Mechanical Engineer to the Clyde Navigation Trust) to preserve and lubricate propeller shafts of self-propelling hopper barges, were given in a paper read by him before the Institution of Mechanical Engineers at the summer meeting of that Institution held in Glasgow from June 25 to 29. Like most large harbour authorities the Clyde Navigation Trust have on service a number of self-propelling hopper barges. Their first hopper barge, in fact, was constructed by William Simons & Co., Renfrew, as long ago as

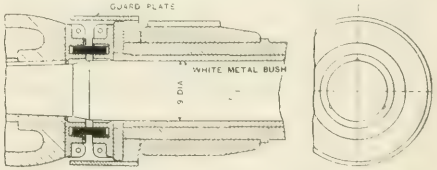
View of Tail Shaft Withdrawn.

1862. This vessel, which is generally recognised as the first sea-going hopper barge, was capable of carrying about 300 tons, and enabled spoil to be deposited at a deep part of the estuary of the river, about 27 miles from Glasgow, and several features in its design are retained in hopper barges of the present day. Of recent years the position of the spoil ground has been altered to 40 miles from Glasgow and the more modern barges are each capable of carrying 1,200 tons of sand. These vessels



Sand Excluding Apparatus for Propeller Shafts. Elevation.

are 205 ft. overall, by 35 ft. moulded breadth, by 15 ft. 6 in. moulded depth, and are propelled by twin-screw triple-expansion steam engines of 1,200 i.h.p. collectively at a speed of 10½ knots when fully loaded. As the vessels are working for a considerable part of the time in water containing much sand in suspension, the appliance referred to by Mr. Fife is of great



Sectional Elevation of the Apparatus.

interest. This appliance, which is fitted to the latest type of barge, four of which were built by Lobnitz & Co. in 1911, is designed to lubricate and preserve the propeller shafts. In actual practice it has greatly reduced the maintenance charges and enabled the vessels to be more fully

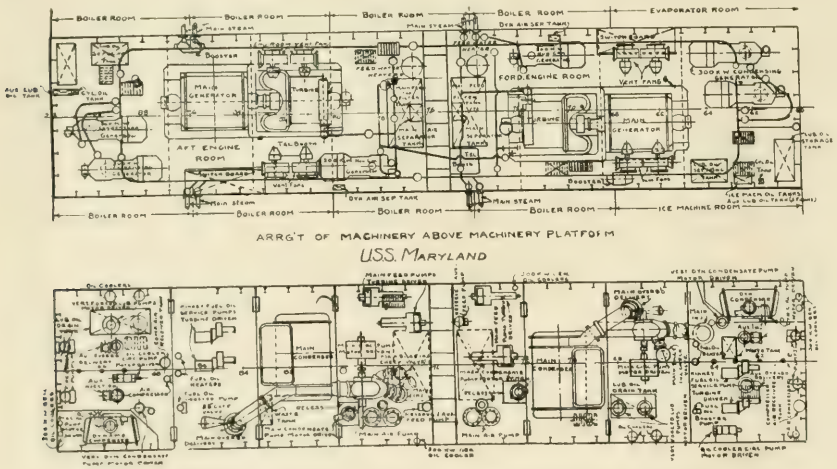
(*Continued on page* 274.)

THE MACHINERY OF THE U.S.S. "MARYLAND."*

Some Interesting Details of the Turbo-Electric Propelling Machinery of the U.S. Battleship "Maryland," manufactured by the General Electric Company, Schenectady, U.S.A.

The U.S.S. *Maryland* was authorised by Act of Congress August 29, 1916, at a contract price, to include hull and machinery, of cost plus fee. The other ships of the *Maryland* class authorised under this Act were the *West Virginia, Colorado* and *Washington*. These ships, together with the *Tennessee* and *California*, whose major difference from the *Maryland* class is in battery alone, were built as a result of the study of the last war, particularly with respect to watertight integrity, arrangement and design of battery, and the use of turbo generators and motors for main propulsion.

The contract for the ship was signed with the Newport News Shipbuilding & Dry Dock Company on December 5, 1916, with supplementary contracts dated May 29, 1917, and June 18, 1920. The General Electric Company, of Schenectady, N.Y., was the sub-contractor for main propelling machinery,



U.S.S. "Maryland." Lay-out of Main Generating Machinery and Auxiliaries.

the six 300 kw. turbo-generating sets and the necessary spare parts. The following are the hull dimensions and particulars :—

Hull Particulars.

Length between perpendiculars	600 ft.
Projection forward of fore perpendicular	24 ft.
Length overall	624 ft.
Length of straight keel	539 ft.
Breadth, moulded	96 ft.
Draught, L.W.L., designed above bottom of keel	30 ft. 6 in.
Displacement, normal, at 30 ft. 6 in. draught	32,630 tons.
At designed draught—	
Area immersed midship section	2,904 sq. ft.
Area, L.W.L. plane	42,430 sq. ft.
Wetted surface	76,611 sq. ft.
Coefficient of fineness—	
Block	·634
Midship section	·977
L.W.L.	·725
Fuel oil, 95 per cent. capacity of compartments	5,000 tons.
Reserve feed water, capacity of compartments	575 tons.
Fresh water tanks, capacity	175 tons.

NOTE.—The displacement includes two-thirds full supply of stores, fuel and water; full supply of ammunition.

The main propelling machinery of the *Maryland* consists of two direct connected turbine-driven alternators, two main condensers, four two-phase induction motors, two booster generating sets for excitation, supplied with power by any one of six 300-kw. geared turbine generating sets, motor and steam-driven auxiliaries, main switching equipment, exciter and auxiliary switchboard, ventilating blowers for the main motors, control panels and wiring for electric auxiliaries, together with necessary wiring for connecting main generators and motors with the switching equipment. The weight of the main propelling machinery as listed above is about 1,800 tons. The general arrangement of machinery is shown herewith.

Main Turbines.

The main turbine arrangement is shown on page 263. The turbine is a General Electric Curtis type, Form W, 10-stage, type HT, 11,000 kw., 2,065 r.p.m., 250 lb. steam pressure. It has 10 wheels, all having a single row of blading except the first, which has two. The steam enters the turbine through the strainer to a 14-in. throttle valve, under the control of an emergency governor, thence to the controlling valves, which in turn are under the control of the operating governor. The quantity of steam admitted is regulated by the operating governor which operates the pilot valve of the hydraulic cylinder by an arrangement of floating levers which are in turn connected to the governor lever. The pilot valve admits oil under pressure to the hydraulic cylinder and thereby moves a piston and rack up or down and thus closes or opens the controlling valves by means of a horizontal camshaft. The steam chest has 10 controlling valves with ports leading from each valve to a group of first stage nozzles. These controlling valves are operated by cams so arranged on the camshaft that the valves are raised or lowered in succession, admitting such quantity of steam as to maintain a predetermined speed.

Lubricating System.

The main lubricating system for the turbine bearings, generator bearings, main motor bearings, main Kingsbury thrust bearings, one which also provides oil for the main hydraulic governor system, covers an arrangement of two distinct systems, one each on the port and starboard side. Pressure is supplied to either of these systems by two motor-driven vertical pumps or, in case of the failure of these pumps, by a

* Abstract of paper read by Lieut.-Commander F. T. Van Auken, U.S.N., before the American Society of Naval Architects, and published in the Journal of that Society.

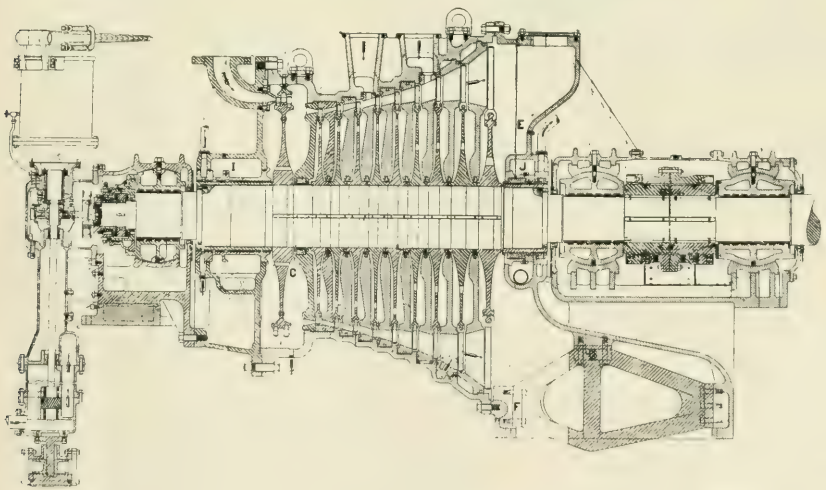
spiral gear oil pump driven directly from the governor end of each main turbine.

The motor-driven pumps are rated at 300 gallons per minute at 1,500 r.p.m. to give 100 lb. pressure, which normally will provide 75 lb. gauge pressure to the system after passing through various bafflers. Oil is supplied to the hydraulic cylinders at 75 to 80 lb. pressure and, by the use of bafflers and needle valves, is throttled down to from 5 to 10 lb. for the bearings.

Main Condensers.

The *Maryland* is equipped with the conventional type of Navy, two-pass, surface condenser, having a total cooling surface of 12,050 sq. ft. The first or lower pass comprises 3,499 tubes and the upper or second pass 3,515 tubes, all the tubes being $\frac{5}{8}$-in. outside diameter and 10 ft. $8\frac{1}{2}$ in. long. Of the tubes included in the lower pass, 438 are in an air-cooling space. The inside diameter of the shell is 9 ft. 3 in. and the length between the tube sheets 10 ft. 6 in. A single exhaust opening is provided on the top of the shell which is bolted to the lower side of the main turbine exhaust hood. A baffle plate is fitted over the air section to cool and condense the vapours. Entirely separate air ejection and condensate suctions are provided, the air being drawn from the uppermost part of the air-cooling space located on the shell side opposite the exhaust opening.

Two separate systems for the removal of air and condensate are provided. The first is a twin beam, wet and dry air pump —Worthington-Blake-Knowles, size 11 × 30 × 27. The second system and the one always in use except in case of breakdown, or for putting a daily vacuum on the condensers, consists of three steam air ejectors arranged in parallel together with a vertical motor-driven condensate pump. The



Assembly of 10-Stage Curtis Steam Turbine for Direct Connection to A.C. Generator.

air ejectors are of the Westinghouse Le Blanc type, two-stage, size G, having the following rating :—

Steam pressure, lb. per sq. in.	125
Pounds of free air per hour at 28-in. vacuum referred to 30-in. barometer	36
Pounds of free air per hour at 28·5-in. vacuum referred to 30-in. barometer	27
Steam consumption, lb. per hour	880

The air ejectors discharge into air separating tanks or auxiliary condensers to condense the steam and to rid the condensate of air. The steam is condensed by the condensate pump discharge and hence the major part of the total heat content of the steam used in the ejectors is regained.

Main Condensate Pumps.

One motor-driven double inlet, volute, vertical, single-stage and single-runner pump is provided for each condenser. The pump suction is taken off the main air pump suction and is discharged to the air separating tank. The complete unit is manufactured by the Westinghouse Electric Manufacturing Company.

The following are some particulars of the pumps :—

Diameter, suction pipe	10 in.
Diameter, discharge pipe	6 in.
Capacity	500 galls. per min.
Head	70 ft.
Motor—	
Horse-power	19
Voltage	240
Speed	1,400/1,700 r.p.m.
Windings, compound, interpole.	

Main Circulating Pump.

One Worthington centrifugal, horizontal, motor-driven, variable speed, double inlet, single-stage, main circulating pump is provided for each main condenser.

PUMP DATA.

Capacity, gallons per min. at 500 r.p.m.	19,000
Head, at 500 r.p.m.	30 ft.
Diameter—	
Suction	28 in.
Discharge	28 in.
Impeller	25·5 in.
Motor windings, shunt.	
Motor—	
General electric.	
Horse-power	125/250
Voltage	115/230
Speed	400/500 r.p.m.

Main Generators.

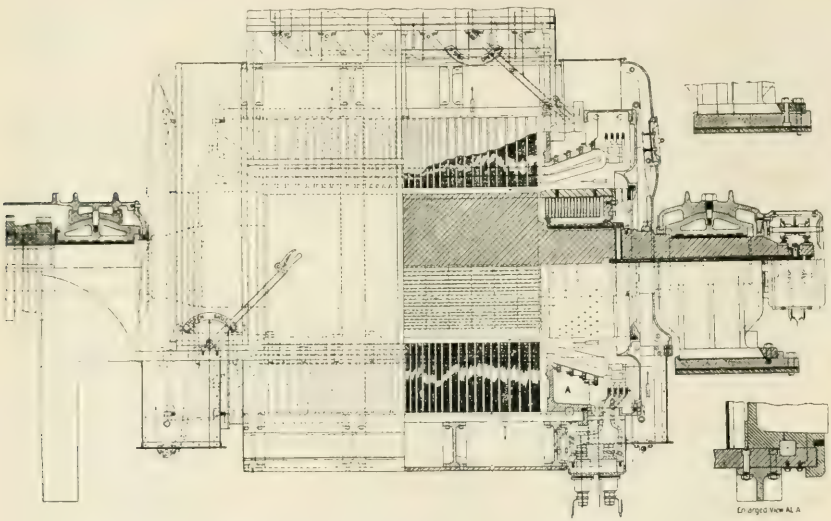
The main generators are General Electric type AQB, Form HT, having a nominal rating of 13,400 k.v.a. at 2.065 r.p.m.; voltage, series connection of stator, 4,325; ampres, 1,550; kw. 11,000; power factor, 82 per cent. Their maximum capacity is approximately 14,800 k.v.a.

The two-phase windings of the stator are so arranged that they may be connected (1) in series or square for high voltage connection and (2) in parallel or diametrical for low voltage connection. The square or high voltage connection is used only when two generators are being used for main propulsion; that is, when the forward generator is connected to the two starboard main motors and the after generator to the port main motors. The diametrical or low voltage connection is used when only one generator is being used to propel the ship, the four main motors being tied in parallel. An interlock is provided so that the high voltage generator connection may not be used on one generator driving all four motors, in order to prevent the overloading possible on one generator under such conditions.

Main Generator Rotor and Field Coils.

The rotor or revolving field consists of a one-piece, heat-treated, solid steel forged shaft serving both as a field yoke

and core. Radial slots, milled in the body portion of the shaft, carry the distributed field winding consisting of several coils per pole, there being only one pair of poles. Each rotor coil turn is formed from copper strip, wound edgewise, insulated with mica tape, and wound into the slot turn after turn. The complete coil is insulated from the rotor slot by a special combination armour and the coil is anchored in the slot by steel and brass wedges. Beyond the slots the coil end portions are insulated, blocked, and held in position by broad nickel steel retaining rings shrunk on to projecting lugs of the rotor body and over centring rings at the outer ends of the retaining rings. Holes are drilled in the retaining rings and a passage left between rings and rotor body for the necessary ventilation for the field coil ends.



Assembly of 11,000 kw., 2,065 r.p.m. A.C. Generator.

The centring rings, besides supporting the retaining rings, also carry the multivane ventilating fans and whatever balancing weights are necessary.

The generator is further equipped with steam-heating coils, located in the end shields at the bottom of the machine, to raise the temperature of the air inside the generator approximately 5 deg. above the room temperature and hence prevent accumulation of moisture on the stator and rotor windings. An additional small heating coil above the terminal board prevents the accumulation of moisture at the terminals.

Fire extinguishing is provided for by the installation of steam coils consisting of circular segments of brass tubing having a number of orifices drilled in them through which steam may be discharged over the windings, these segments being fastened to the fore and aft end shields at a position opposite the stator coils. Both inlet and outlet air dampers are fitted on the ventilation system and these dampers would be closed in case of fire.

The rotor is provided with bearings of the pedestal type, self aligning, fitted for forced lubrication. Provision is also made so that the air gap adjustment may be accurately made by shims. The pedestal on the collector end is insulated from the foundation to prevent circulating stray field currents damaging the bearings.

Main generator excitation is provided from any one of six 300-kw. direct current, three-wire, 120/240 volts turbo-generating sets, there being three of these sets in each engine room, two being condensing sets and one non-condensing. The non-condensing sets are normally used to provide both the excitation power and that for the main propulsion motor-driven auxiliaries. Two motor generator booster sets are also provided, one in each engine room, for giving wide variation to the excitation voltage.

Main Motors.

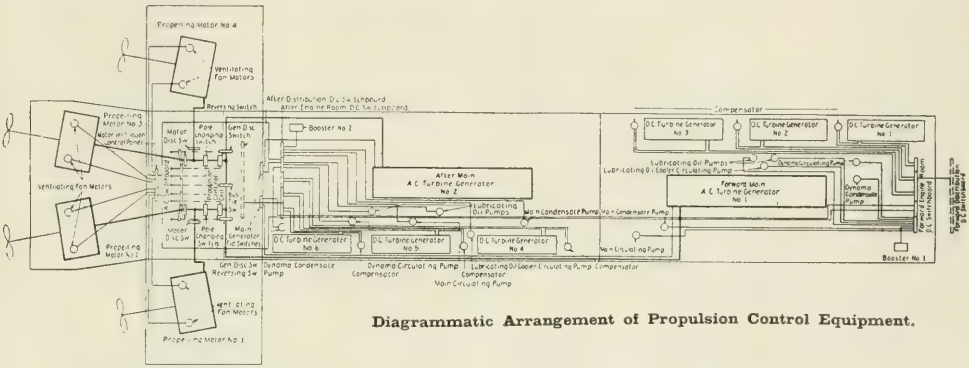
There are four two-phase induction motors, each direct connected to one propeller shaft, the thrust of which is taken by a Kingsbury thrust bearing placed between the motor and propeller. These motors have a nominal rating of 7,000 s.h.p. each when running at a speed of 170 r.p.m. at an applied voltage of 4,400 with the generators square or series connected. The main motors are wound quarter phase with a primary winding in the stator so arranged that the number of poles can be changed from 24 to 36. Since the generator has but two poles the revolutions per minute of the motor will approximate respectively 1/12 or 1/18 of the revolutions per minute of the generator. Up to and including 15 knots, therefore, or 118 r.p.m. of the main motors, the 36-pole connection is used—corresponding to a turbine speed of 2,170 r.p.m., all intermediate speed changes being made by the turbine governor. Beyond 15 knots to maximum speed, the 24-pole connection of the motor stators is used. For instance, at 15 knots the motor speed of 118 r.p.m.—on the 24-pole connection—corresponds to a turbine speed of 1,440 r.p.m. All intermediate changes up to maximum speed are made through the turbine governor. Thus, within the propeller speed range is included two points at which the turbine is running at maximum speed and hence best economical condition, a feature largely accounting for the high efficiency of electric propulsion. Furthermore, one generator alone is used up to 17 knots, the motor pole connections being changed at 15 knots from 36 to 24. The experience of the *Maryland* has further shown that in an emergency, one generator can assume a greater load than that for 17 knots, a feature which might well become paramount in action.

Main Propulsion Auxiliaries.

The following auxiliary equipment necessary for the operation of the main generators and motors is installed :—

- 2 300-kw. non-condensing geared turbo-generator sets.
- 4 300-kw. condensing geared turbo-generator sets.
- 2 19,000 g.p.m. motor-driven main circulating pumps.
- 2 1,000 g.p.m. motor-driven dynamo circulating pumps.
- 4 80/250 g.p.m. motor-driven lubricating oil pumps.
- 8 10,000 cub. ft. per minute motor-driven main motor blowers.
- 2 oil cooler circulating pumps, motor driven.
- 2 18-kw. 60-volt motor generator boosters.

The above pumps are all motor driven and controlled from the D.C. engine room switchboards by General Electric automatic control panels of the CR type. The ventilating blower motors are controlled by switches on the main motor ventilation control panel in the main control room through hand-operated control panels of the CR type at the motor itself. The booster motor is started by means of a pump type switch mounted on the D.C. engine room switchboard.


Propelling Motor No 4
Ventilating Fan Motors
Reversing Switch
After Distribution Board Starboard
After Engine Room D.C. Switchgear
Compensator
Propelling Motor No 3
Motor Unit Equipment
Motor Dist Sw
Gen D.C. Switch
Booster No 2
D.C. Turbine Generator No 3
D.C. Turbine Generator No 2
D.C. Turbine Generator No 1
Lubricating Oil Pumps
Lubricating Oil Cooler Circulating Pump
Ventilating Fan Motors
After Main A.C. Turbine Generator No 2
Forward Main A.C. Turbine Generator No 1
Dynamo Condensate Pump
Forward Engine Room Switchboard
Main Condenser Pump
No 4 Condenser Pump
Main Circulating Pump
Booster No 1
Propelling Motor No 2
Main Generator Field Switches
D.C. Turbine Generator No 6
D.C. Turbine Generator No 5
D.C. Turbine Generator No 4
Gen Dist Sw Reversing Sw
Dynamo Condensate Pump
Dynamo Circulating Pump Compensator
Lubricating Oil Cooler Circulating Pump Compensator
Main Circulating Pump
Propelling Motor No 1

Diagrammatic Arrangement of Propulsion Control Equipment.

Shafting and Bearings.

There are four lines of shafting each driven by the main propelling motors, each line of shafting being composed of the following number of separate sections all rigidly coupled together :—

	Inboard shafts.	Outboard shafts.
Thrust shaft	1	1
Line shaft	2	2
Stern tube shaft	1	1
Propeller shaft	1	1

Thrust Shafts.—Thrust shafts are fitted with thrust bearings of the Kingsbury type, having a steady bearing near the after coupling and at each end of the thrust bearing. Coupling discs and thrust collar are forged integral with the shaft.

Line Shafts.—These are hollow forged with disc couplings at each end forged integral with the shaft. The after section of the outboard line shafting and both sections of the inboard line shafting are carried in two steady bearings, one near each end of the shaft. The forward section of the outboard line shafting is carried by a single steady bearing just forward of its centre.

Stern Tube Shafts.—These are hollow forged and secured to the line shafts through special disc couplings so that the shaft may be removed through the stern tube bearings. Composition casings are shrunk on the shaft where they pass through stern tube and stern tube bearings. Special sleeve couplings are fitted to secure the stern tube shafting. Lignum vitæ bearings are provided. A standard stuffing box gland is provided where shaft passes through the forward bulkhead at frame 115 for the outboard shafts and frame 124 for the inboard shafts. Fairways are fitted where the shaft leaves the stern tube bearings.

Propeller Shafts.—These are hollow forged, taper turned at after end to suit propeller hub and have composition casing shrunk on where they pass through the strut bearing, each shaft being carried by a single lignum vitæ line strut bearing. Fairways are fitted at each end of this bearing.

Inboard Couplings.—The stern tube shaft is upset at its inner end and has fitted over this upset portion a sleeve secured by four keys. At rear of the sleeve a collar, made in halves, is fitted against the shoulder formed by the upsetting, these collars being secured to the sleeve and line shaft disc by eight through bolts.

Outward Couplings.—These couplings are of the solid sleeve type, taper bored for the reception of the shafts, secured to the shafts by two feather keys (axial) and one cross key (radial).

Where line and thrust shafts pass through watertight bulkheads a special type sleeve stuffing box is provided. A sleeve with standard type stuffing box gland fitted in its end and supported from the shaft foundation is fitted in the bulkhead. By this arrangement bulkhead deflection may occur without binding the shaft.

Shaft Data.

Thrust shafts—	
Length, inboard	22 ft. 2 in.
Length, outboard	22 ft. 2 in.
Diameter	15¼ in.
Diameter at bearings	15½ in.
Diameter, axial hole	7⅝ in.
Thrust collar, number	1
Thrust collar, thickness	6½ in.
Thrust collar, outside diameter	30⅜ in.
Thrust collar, inside diameter	15·74 in.
Line shafts—	
Forward, inboard, length	28 ft. 8¾ in.
After, inboard, length	28 ft. 8¾ in.
Forward, outboard, length	19 ft. 8 in.
After, outboard, length	19 ft. 8 in.
Diameter	15¼ in.
Diameter at bearings	15½ in.
Diameter, axial hole	7⅞ in.
Stern tube shafts—	
Length	41 ft. 9 in.
Diameter	16⅞ in.
Diameter at couplings	18⅜ in.
Diameter at bearings	18⅜ in.
Diameter, axial hole	8⅜ in.
Diameter, axial hole at outboard coupling sleeve	4 in.
Sleeve, thickness at bearings	⅝ in.
Taper in outboard couplings	1 in. per foot.
Propeller shafts, outboard, length..	34 ft.
Propeller shafts, inboard, length	31 ft.
Diameter	16⅞ in.
Diameter, axial hole	8⅜ in.
Diameter, axial hole at propeller hub and outboard coupling	4 in.
Taper in outboard coupling	1 in. per foot.
Taper in propeller hub	1 in. per foot.
Coupling discs—	
Diameter	22½ in.
Number, bolts	8
Inboard couplings—	
Diameter of sleeve, outside	30 in.
Diameter of sleeve, inside	18⅜ in.
Length of sleeve	13 in.
Thickness of collar	3½ in.
Coupling bolts—	
Number, each coupling	8
Diameter (taper) at face of coupling	2¾ in.
Outboard coupling—	
Length of sleeve	63½ in.
Diameter, outside	22½ in.

Steaming Trial Data, U.S.S. *Maryland*.

	12 Hour. Full Power.		4 Hour. 19 Knots.		4 Hour. 15 Knots.		4 Hour. 10 Knots.	
	A.	B.	A.	B.	A.	B.	A.	B.
Average speed, knots	21·07	21·07	18·97	18·97	14·98	14·98	10·07	10·07
Displacement, tons	32,655	32,655	32,550	32,550	32,615	32,615	32,518	32,518
Average s.h.p.	29,200	29,750	19,000	19,470	9,190	9,225	2,575	2,560
Total oil consumption per hour, lbs. (19,500 B.T.U.)	28,651·6	28,710·5	20,857·9	20,115·9	10,986·1	11,223·7	4,912·8	4,524·3
Oil consumption per s.h.p., lbs. per hour	·982	·966	1·064	1·002	1·195	1·216	1·902	1·762
Oil consumed per square feet H.S., lbs. per hour	·614	·625	·518	·500	·477	·484	·427	·393
Oil consumption per knot, lbs. (19,500)	-	1,339	—	1,022	733·9	748·7	—	446
Water consumption, lbs. per s.h.p. per hour	10·817	11·555	10·775	—	10·915	11·066	13·20	14·56
Number of boilers used during run	8	8	7	7	4	4	2	2
Steam at boilers, lbs. gauge	250	250	260	260	265	265	250	250
Steam at turbine, lbs. gauge	246	246	255	255	258·8	260	241·3	243·0
Steam at turbine, temperature degrees Fahr.	436·7	436	—	—	—	—	424	424
Vacuum, inches	29·0	29·0	28·5	28·5	28·0	28·0	28·0	28·0
Barometer, inches	29·98	29·98	30·22	30·22	30·22	30·22	30·20	30·20
Fireroom, air pressure, inches of water	5·5	5·5	4·8	4·8	3·7	3·7	2·8	2·8
Shaft r.p.m.	171·24	171·24	150·74	150·74	118·01	118·01	78·01	78·01

Column A—Auxiliary exhaust into dynamo condenser. Column B—Auxiliary exhaust into main turbines.

Main Control Equipment.

The main control equipment consists of the following:—

(*a*) Two D.C. engine room switchboards each of which consists of five panels, one each for each 300-kw. generator and two for controlling the motor generator booster and the 10 engine room auxiliary motors together with the main motor ventilation supply.

(*b*) One motor ventilation control panel supplying the main propulsion blower motors and the constant voltage D.C. control circuit for energising all magnetic locks, contactors, and indicating lamps.

(*c*) One alternating current switchboard consisting of a structural steel cell with equipment for controlling both main generators and the four main motors.

Refrigerating System.

The ice machine room is situated between frames 60 and 68 on the starboard side of the platform deck and has installed therein three ice machines of the CO_2 type with a capacity of 3 tons each, the compressors of which are driven by reciprocating steam engines; all of the equipment is supplied by Kroeschell Brothers Ice Machine Company. Two Worthington-Blake-Knowles simplex brine pumps, size 6 in. × $5\frac{3}{4}$ in. × 6 in., are installed for brine circulation. Two exactly similar pumps are installed for sea-water circulation. There are also fitted three condensers for the CO_2 gas; one brine storage tank; one ice-making tank of three compartments; two brine cooled scuttle-butts, one each located on main and second deck; cold storage room for meats, eggs and cheese, cream and butter, and cooling room for officers' mess, all located on third deck, and a 40-quart ice cream freezer and hardening box located on main deck.

Evaporator Plant.

As originally installed this plant was of the high pressure type. It consists of three first effect evaporators, three second effect evaporators, three-coil drain feed water heaters, two vapour feed water heaters, four distillers, two evaporator feed pumps, two fresh water pumps, two distiller circulating pumps, one brine pump, one distiller air pump and a Cochrane precision meter to measure the output. All this equipment was located in the evaporator space, between frames 60 and 68 on the first and second platforms. The above is now being replaced with a low pressure installation built at the Navy Yard, New York; it is designed to operate on exhaust steam and should effect a very considerable improvement in economy.

The thrust bearings, propellers, communication system, general piping arrangement, boilers, boiler control, and fuel oil system are practically identical with the U.S.S. *Tennessee*.

The final official trials of the U.S.S. *Maryland* were held November 14-17, 1921, off Rockland, Maine, and *en route* Rockland to New York. Just before the trials the ship was docked at the Navy Yard, New York. The ship averaged 21·77 knots on five high speed runs at a displacement of 32,500 tons, developing an average shaft horse-power at this speed of 36,167 with a maximum of 36,544. The trials were most successful, and unattended with a casualty of any kind. There is appended above in tabular form the principal data obtained on trials.

CONDENSER TUBE CORROSION.*

By R. Le BOYSSON,
Ingénieur principal de la marine française.

Trouble has been experienced with condensers in the merchant service, the navy and in large power stations ashore. The most serious trouble is unquestionably that of the tube corrosion due to the action of the circulating water and especially when this is salt water. As a good condenser is essential to the successful working of the machinery and evaporators, this question is engaging the most serious attention of constructing and operating engineers. In all countries attempts are being made to better the situation and considerable research work is being carried out. To the best of the writer's knowledge, however, the question has never been methodically studied in France. In Germany a few experiments have been made by Lasche but the most comprehensive work has been done in England and Switzerland. In England, the Corrosion Research Committee of the Institute of Metals was especially formed to study the corrosion of metals and particularly the action of salt water on brass tubes. This committee, under the chairmanship of Mr. Bengough, has collated many observations and carried out numerous experiments. The chief object of the committee was to discover which conditions promote corrosion and consequently further to discover adequate means of protection against corrosion. The committee has thus been led rather to disentangle the numerous secondary factors of the phenomenon than to attempt the penetration of its internal mechanism. It has formulated a number of rules which should be observed in practice in order to increase the resistance of tubes to corrosion. The various reports presented to the Institute of Metals were summarised by them in a pamphlet published last year. In Switzerland, M. de Wurstemberger has made some notable laboratory experiments in which he has attempted mainly to penetrate the internal mechanism of corrosion and its fundamental causes. A preliminary report appeared in the *Revue Polytechnique Suisse*, of October, 1919, and the complete results were published in the *Revue de Metallurgie* for November, 1921. The English and Swiss works are thus on entirely different lines. They result in rather different conclusions being drawn from them, some of which are apparently contradictory. We shall now examine the two sets of experiments.

Resumé of the English Recommendations.†

Bengough gives the following rules for the design of condensers :—

1. For tubes and washers a Cu 70, Zn 29, Sn 1, alloy should be used with no more than ·1 per cent. Fe present. The tube should be exposed for three hours at a temperature of 350° Fahr. ± 25° Fahr. to an oxidising atmosphere.

2. Corrosion once commenced cannot be stopped. New tubes should be cleaned, although with old tubes this is not so necessary.

3. When iron protection slabs are made use of, these should be changed at the same time as the tubes. Soft metal rings should be placed in the stuffing boxes.

4. Electrolytic protection from an external source should be tried if the cases of damage are excessive. The current should be used continuously and not intermittently. The installation should be carefully constructed and reversals of polarity avoided.

5. Leaky fittings may lead to corrosion.

6. Always place the steam pipes at the top.

7. Avoid too high a water speed (2 metres per second at the most) as well as variations in speed. Avoid mixtures of air, oil, &c.

8. Avoid exposing new tubes to the action of doubtful water (that of certain river estuaries for example).

Thus it can be seen that the English experiments did not attempt to penetrate the internal mechanism of corrosion. This latter is considered as purely a chemical phenomenon of which the process has not been cleared up. The experimenters have chiefly sought to discover the influence of certain practical factors. A number of empirical rules have been thus established which permit of the amelioration of the conditions of working. These rules are very interesting and should be remembered in the construction and operating of condensers. But it is very evident that they are only palliatives of limited utility since the very fundamentals of the subject are left untouched.

Swiss Experiments.

M. de Wurstemberger commences by showing that selective corrosion and de-zincification are two different phenomena. Indeed, selective corrosion can take place by local action on pure copper. On brass tubes two types of attack are found : local selective with disolution of the two constituents and de-zincification with dissolution of the zinc only, also a deposit of spongy copper.

1. *Selective Corrosion.*—Selective corrosion is an electrolytic phenomenon due to the presence of partly soluble salts forming local differences of potential ; the relevant factors being the current, the nature of the alloy, the nature of the electrolyte and the *régime* of the apparatus. M. de Wurstemberger has made numerous experiments which have given the following results :—

(*a*) Selective corrosion is assisted by basicity.
(*b*) Salt water, especially that which is stagnant and which has been in prolonged contact with particular metals, becomes basic.
(*c*) The electric reaction of the metal does not only depend upon the metal itself but also upon the properties of the deposited salts.
(*d*) Secondary anodic phenomena are particularly important.
(*e*) Metals of several valencies yield in the first place salts of a lower valency, which are transformed by oxydisation into salts of a higher valency.
(*g*) Selective corrosion exists on pure copper. This metal is the principal agent of the phenomenon which is essentially electrolytic.
(*f*) Currents due to the presence of metals forming in the condensate are particularly destructive. In fact, differences of potential might be thereby produced which result in tubes becoming the anode in the circulating water.

The author particular insists on this point and shows the danger of the presence of metals of different qualities (copper and iron, for example) in vacuum chambers. He adds that the largest condensers are the most exposed to corrosion because of the greater probability of leaks.

2. *De-zincification.*—De-zincification is first of all produced by the complete dissolution of the brass and the liquid being charged with zinc oxychloride and copper oxychloride becoming basic. Then the copper re-deposits itself in a spongy state. Brasses are ordinarily a mixture of two different alloys : small polyhedral β crystals of Cu 52, Zn 48 disseminated in a mass of large α crystals Cu 70, Zn 30. The β crystals are much more attackable and are the first to be destroyed. De-zincification does not commence spontaneously ; it is always

* Abridged translation of an article which appears in the June number of the *Bulletin Technique du Bureau Veritas.*

† In the original article the author gives a very complete account of the English experiments. As these are already familiar to our readers we only reprint here the author's résumé of this work.

set up by a selective corrosion which produces a local concentration of copper salts, generally cuprous chloride produced by the electrolysis. The phenomenon presents two different aspects: in a mono-phase alloy 70, 30 only containing α crystals, mass de-zincification takes place; in a bi-phase alloy containing α and β crystals de-zincification takes place by ramification following the boundaries of the β crystals. The author distinguishes selective corrosion from de-zincification as follows: Selective corrosion is the result of a local electrolysis produced by extraneous or local currents; it only progresses if the current persists. De-zincification is an action attached to selective corrosion either progressing or even stopped. Once started the phenomenon progresses by itself with the appearance of local electrolysis. The starting is produced by a concentration of electrolysed copper salts, then the brass dissolves in the copper salt solution, next the copper separates out and finally is deposited. The operation is facilitated by this deposit of copper which acts as a depolariser.

Experimental results show that an alloy of less than 70 per cent. Cu is rapidly destroyed by the de-zincification of the β crystals. A 70 per cent. Cu alloy with α crystals resists de-zincification better; the presence of 1 per cent. Sn facilitates the formation of α crystals. An alloy of greater than 70 per cent. Cu whilst less conducive to de-zincification is much more sensitive to selective corrosion by local currents. M. de Wurstemberger next takes up the conclusions of the Corrosion Research Committee and questions them because of the Committee's considering that corrosion is a chemical and not an electrolytic phenomenon.

3. *Protection.*—M. de Wurstemberger does not believe in the efficacy of cleaning the tubes proposed by the Corrosion Research Committee, the origin of the phenomenon being microscopic. He proposes the following:—

1. Suppress extraneous currents by short-circuiting them. This applies particularly to land condensers subject to the action of stray currents.

2. Suppress the currents arising in the condensate. As the absolute neutrality of this cannot be guaranteed it is essential that the condenser air chamber be made of only one quality of metal. It is also essential to ensure electrical contact between the tubes and the end plates so that the current produced has a complete circuit through metallic conductors.

3. Use an alloy 70, 30 or 70, 29, 1 with suitable heat treatment and only having large α crystals. This does not signify that its structure should be coarse grained, α crystals being really fine grained. Tinning is dangerous for it gives rise to selective corrosion.

4. Electrolytic protection can be useful if the cathodic action can be spread uniformly over the surface to be protected. This condition is realised to some extent in boilers but in condensers it seems impossible of realisation. The presence of secondary cathodic products having an alkaline reaction is clearly harmful. Moreover the variable state of the electrodes of the tubes varies the radius of action of the protection; there thus exists a zone which is alternatively anodic and cathodic, and particularly dangerous.

For these reasons M. de Wurstemberger does not endorse electrolytic protection.

5. The most harmful elements are deposits of copper, protections of variable intensity (zinc and iron plates, &c.) and dissolved chlorides.

By applying these observations condensers can be constructed which are likely to resist corrosion. But M. de Wurstemberger considers that the most rational process is that of an electrolytic protection which produces in a continuous manner the elements necessary for the correction of the alkalinity of the salt water. In other words the water must be electrolysed and must pass into the condenser only, the anodic acid products. This process has resulted in several patents and Messrs Escher Wyss are now constructing condensers on this system.

Lasche's Experiments.

Lasche* has proposed the following classification of corrosion: (1) Corrosion by external currents; (2) Corrosion by differences of potential between neighbouring parts of the same tube; (3) Corrosion by foreign bodies forming couples (coal for example); (4) Corrosion by purely chemical agency. He omits from his classification corrosion due to differences of potential arising from the condensate. On the other hand cause (3) above does not seem to be very important. As a means of protection Lasche only proposes electrolytic protection.

Discussion of Results.

The English and Swiss experiments lead to rather different conceptions of the corrosion of condensers. The Swiss experiments being carried out with considerable care and precision appear to be the more comprehensive, and from them one can conclude that corrosion is not a chemical phenomenon but one of local electrolysis. Consequently the recommendations of Bengough can only be considered as palliatives. Only the correction of alkalinity can give a complete solution; and it remains to be seen whether it can be realised in a practical manner. However, the points upon which English and Swiss experiments are agreed ought to be borne in mind. Certain details should be considered; for instance the influences of the galvanic effects of the condensate and that of the temperature. Experience has shown in general, that auxiliary condensers which are generally rather hot do not resist corrosion well. It has been observed that condensers kept in reserve aboard ship, but not dry, suffer very quickly from corrosion, the cause being undoubtedly the alkalinity of the stagnant water.

The observance of the following rules is suggested therefore:—

1. *Nature of Metal.*—The condensers should be constructed throughout of the same metal as the tubes. The best metal in ordinary cases is brass (Cu 70, Zn 29, Sn 1) exposed for three hours at a temperature of 350° Fahr. ± 25° Fahr. to an oxidising atmosphere. The iron contents should not exceed 0·1 per cent. Under microscopic examination the metal should only show large α crystals.

2. *Stuffing Boxes.*—Stuffing boxes should be fitted with a soft metallic ring at the bottom to ensure electric contact between the tubes and the plates. The circulating water pipes may be an important source of corrosion and hitherto unexplained troubles may be due to this cause; it is thus of the greatest importance to have the condensers perfectly tight.†

3. *Circulation.*—The cooling surface should be liberally designed and the circulating water should only pass through the condenser in one direction without multiple circuits and with spring chucks to ensure a uniform distribution. The velocity of the circulating water should not exceed 2 m/sec. It is better to keep it between 1·3 and 1·5 m/sec. Air suction pipes for the pumps should be avoided and the water should be drained before entering the condenser. Condenser heating is always dangerous and should be guarded against. The water inlet and ejection valves should be so disposed as to avoid circulation in a closed circuit.‡ All waste matter should be kept out of the circulating water.

* Konstruction und Material in Bau von Dampfturbinen and Turbodynamos. Berlin, 1920.

† The tightness of the condenser is also of course of fundamental importance for the good functioning of boilers and turbines. The question arises whether stuffing boxes should not be dispensed with and the ends of the tubes expanded instead. This has long been the practice in the French Navy in torpedo boats with condensers of beaded tubes with non-parallel end plates. The question could be taken up for straight tubes of parallel plates. If the condenser is of the same metal as the tubes the differences of expansion are small and could be taken up by the elasticity of the tubes which being very long could easily bend without any danger.

‡ This desideratum is generally realised when the ship is going ahead. It also takes place when moored and even when going astern, particularly with turbines where going astern necessitates the use of large quantities of steam.

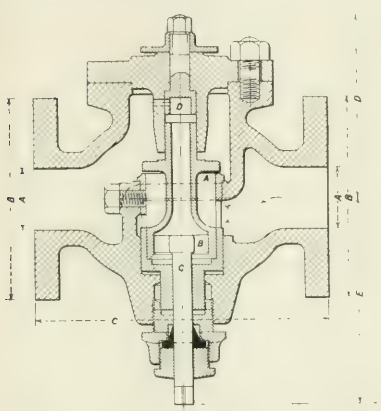
(*Continued on page* 270.)

CLARK'S REDUCING VALVE (EYRE'S PATENT.)

A Description of its Characteristics.

The need of an efficient reducing valve has been long admitted and many attempts have been made to produce a satisfactory article. Types embodying springs and diaphragms are in general use at the present time, but in this class of valve the diaphragm is often a serious drawback. All engineers will confirm that its frequent renewal entails inconveniences and interruptions, resulting in serious loss. Moreover the efficiency of these types in service leaves much to be desired, wire-drawing with its resultant cutting of valve faces and a restricted supply of reduced steam being the most noticeable faults. The desire to overcome these objections and to produce a valve which could be relied upon to give better all-round service, led George Clark & Sons, Hull, the makers, to consider various designs. After considerable experimental work they realised that it would be essential to eliminate springs and diaphragms to achieve the results desired.

In service these valves have proved successful, and in one case where a valve opened out for examination after two years' hard service at sea, the seats and faces were found to be in perfect condition, and during this period it had given every satisfaction without attention. The valve has some special advantages, amongst which the following are important. The valves may be used for all purposes where a reduction of pressure is required either with saturated or superheated steam or compressed air, and it will meet any condition of pressures, reduced or initial, without any alteration of any part beyond regulating to working conditions.



A	B	C	D	E
1"	4 3/32	6"	4 3/16	5 7/16
1 1/4	5 3/16	6"	4 7/16	5 7/16
1 1/2	5 3/32	6 1/2	4 7/8	6"
2"	6 1/2	9 1/2	5 3/4	7
2 1/2	7 5/8	11"	6 7/8	8 1/4
3"	8"	12"	7 3/4	10 1/2
3 1/2	9 1/2	13	8	11
4"	10"	14	9	12
4 1/2	10"	16	10 1/4	13 1/2
5"	11"	16	10 1/2	13 1/2
5 1/2	12"	18	11 1/2	15
6"	12	18	11 1/2	15

The Eyre Patent Reducing Valve. Sectional Elevation and Details.

The internal arrangement is ingenious, consisting of a valve and piston in one casting. Steam is allowed to leak past the piston in small quantities, and this leakage of steam passes through the piston up to a small chamber in the cover, the outlet from this chamber being led to the reduced side of the valve through a cock. The pressure accruing from this leakage constitutes the load, and is regulated by opening or closing the cock on the cover, until the desired reduced pressure is reached. By this arrangement a great range of reduction is obtained. We show in the preceding column a sectional elevation and details of the valve.

Full Bore and Maintenance of Reduced Pressure.—The lift of the valve is not restricted in any way, and automatically adapts itself to any requirements. It maintains a constant reduced pressure regardless of the volume of steam used and is full bore throughout.

Steam Tightness.—This is a feature to which considerable attention has been devoted. The valve is of the double-beat type and does not rely upon accumulation of steam above the reduced pressure to close it, as the effect of the spindle being carried outside the valve body ensures that sufficient load is obtained to keep the valve steam-tight. The difficulty of unequal expansion and distortion of the valve seats is overcome by fitting a cage which is secured at the top only and therefore free to expand uniformly with the valve.

Wire Drawing.—There is a complete absence of wire drawing and its consequent cutting of valve faces. The construction and action of the valve are such that chattering is entirely eliminated.

Upkeep.—Owing to the small number of working parts it will be evident that upkeep is trifling over long periods.

Superheated Steam.—The valve is pre-eminently suitable for use with superheated steam owing to the absence of materials which are affected by extreme temperatures.

Adjustment and Fitting.—An alteration in the reduced pressure can be obtained by merely altering the small cock at the top of the reducing valve. The space occupied by the valve is considerably less than that occupied by valves of other make, it is lighter in weight and will work equally well in a horizontal or vertical position.

Design and Specification.—Valves up to and including 1½ in. bore are made of gunmetal throughout. Above 1½ in. bore the valve bodies and covers are made of cast-iron and the internal parts of gunmetal. When valves are required for use with superheated steam the bodies and covers are made of cast-steel and the internal parts of nickel alloy. The flanges are made to B.S. Table No. 2 and are faced and turned on diameters. The regulating cock in the cover of the valve is made of mild steel, a brass index quadrant being mounted on top. Bosses are provided for pressure gauge cocks, and are drilled to requirements. Suitable brackets for attachment to bulkheads are provided when necessary.

The method of operation of the valve is as follows :—High-pressure steam enters in the direction indicated, causing the combined valve " A " and piston " B " to vibrate. Steam is thus permitted to leak past piston " B " to the inside of com-

bined valve, where its passage to the reduced side is controlled by outlet cock "D." When steam is being used this controlled steam is adjusted by means of cock "D" to give the required working pressure. This controlled pressure being slightly above the required working pressure, and the combined valve and piston are equally loaded on the high-pressure side. When the upward force in the inside of hollow valve is in excess of the combined downward forces on top of valve "A" and on spindle "C" the combined valve is forced upwards until just sufficient steam passes through the valve "A" to give the required reduced pressure on outlet side. The valve commences to close when the pressure on the reduced side approaches the regulated pressure inside the hollow valve. When these pressures balance the valve is shut off tight with a load equal to the area of spindle-controlled pressure and remains shut until the pressure on the reduced side falls sufficiently to remove the load off the valve. When steam is not being used the pressure on the reduced side adjusts itself until the downward force is sufficient to shut the valve tight. This pressure in most cases is much below the working reduced pressure except for very great reductions.

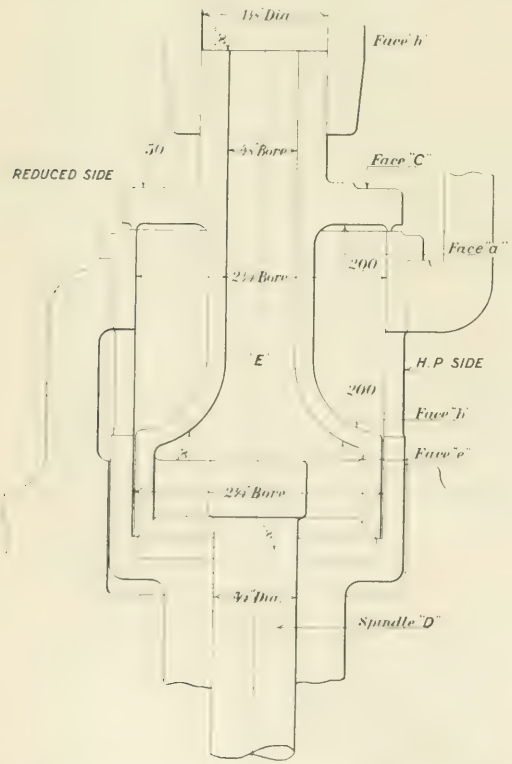


Diagram illustrating the Calculation shown opposite.

The valve can only be adjusted when steam is being used. To increase the reduced pressure slacken the nut on top of the quadrant and turn the quadrant in the direction of arrow until desired pressure is reached. To lower the reduced pressure turn the quadrant in the opposite direction. After the required pressure is reached the nut should be tightened up so as to prevent any movement of the quadrant. It is essential that spindle D is free to move, care should be therefore taken to ensure that the bottom gland is not screwed up too tightly, it is sufficient to tighten this gland by hand. We append below a specimen calculation for the determination of pressure in the chamber E. Data:—

Bore of valve 2-in., boiler pressure 200 lb. per sq. in., reduced pressure 50 lb. per sq. in.

Let the pressure required in chamber E be x lb. per sq. in.

On h.p. side faces a and b are of equal area and therefore balance each other.

Downward force on reduced side = Area of annular face C × reduced pressure.
= $2\frac{1}{4}$ in. × $1\frac{1}{8}$ in. diam.
= 2·98 sq. in. × 50 lb.
= 149 lb. app.

Downward force in chamber E = Area of annular face h × x lb.
− Area of spindle D × x lb.
= $1\frac{1}{8}$ in. × $\frac{5}{8}$ in. diam.
= 1·13 sq. in. × x lb.

Upward force in chamber E = Area of annular face e × x lb.
= $2\frac{1}{4}$ in. × $\frac{5}{8}$ in. diam.
= 3·67 sq. in. × x lb.

When in equilibrium the upward forces must equal the downward forces.

3·67 sq. in. × x = 149 lb. + 1·13 sq. in. × x lb.
2·54 sq. in. × x = 149 lb.
x = 58·66 lb.

The pressure inside chamber E is therefore 8·66 lb. per sq. in. above the reduced pressure when working. It will thus be observed that a very slight increase in the reduced pressure destroys the balance and results in the valve A moving slightly downwards, the effect is opposite when a very slight reduction in pressure occurs. When steam is not being used the pressure on the reduced side immediately adjusts itself until the downward force is sufficient to shut the valves tight. Except for very great reductions the pressure at which the valves close is much below the working reduced pressure.

CONDENSER TUBE CORROSION.—(*Continued from p. 268.*)

The majority of the above proposals are not new and certain are in current practice. They are not, however, all-sufficing and more research requires to be carried out to discover better means of protection. The depositing of protective salts suggested by Bengough may one day be adopted in practice, although the solution to the trouble does not lie in this direction. The use of Benedict metal tubes (Cu 80, Ni 20) offers interesting research. The experiments ought to be made on a sufficiently large scale to be conclusive. We believe that nickel alloys will give better results than present-day brasses; their high price should be counterbalanced by their longer life.

The "Escher Wyss" condensers constructed to M. de Wurstemberger's ideas may be used for marine purposes. If the continuous water alkalinity correction system is definitely shown to be practical it seems that the problem of protection against corrosion may be regarded as completely solved.

A New Beardmore Engine.—William Beardmore & Sons Ltd. have carried out trials on a new type of oil engine. The engine, which is to be fitted in a vessel 710 ft. length overall, now building, is of a six-cylinder type developing 600 b.h.p. at 185 r.p.m. The cylinders are 18-in. bore and 24-in. stroke, and the engine is of the compression port scavenging type. Cylinder heads are water cooled and there is forced lubrication to all bearings including the pump gear.

NOTES FROM THE FOREIGN PRESS.

Specially abstracted for "The Marine Engineer and Naval Architect."

The Torsional Oscillation of Shafts.

The complex phenomena of torsional oscillations cannot be reduced to and solved by any simple formulæ. It is always necessary to examine in very close detail the elastic properties of the vibrating system. However, the problem can be approximately solved for design purposes by a chart which the author has produced based upon Dr. Tolle's development of the graphic method of Dr. Gümbel. The use of this chart involves the preliminary calculation of the "elastic lengths" and elastic masses of the oscillating system. This calculation is dealt with very thoroughly in the first portion of the article under review. The "elastic lengths" of shafts, flange couplings and crank shafts is dealt with. The formula for crank shafts is based upon the work and proposals of Dr. Geiger and Dr. Gümbel. However, the author recommends that his proposed formula should be checked against the actual performance of similar shafts. Next the calculation of mass moment of inertia is gone into the question of connecting rod, fly wheel and propeller being especially considered. Reference is made to a previous article by the author in which the experimental determination of mass inertia was very thoroughly treated. This article appeared in Werft Reederei, Hafen, August 7, 1922, and was referred to in our October, 1922 notes.

In a subsequent article the author proposes to publish the previously referred to chart worked out for a particular type of engine and will compare its results with those obtained from a complete investigation. It will be shown that in many cases the use of the chart will greatly facilitate the calculation of torsional oscillations and especially for internal-combustion engines having several cylinders. The article contains many useful references to the latest works on the subject.—*Dr. C. v.d. Steinen, Werft, Reederei, Hafen*, June 7, 1923, p. 307; 3 pp.; 2 ff.

A New Type of Cargo Mast.

The important fact to remember in the design of a ship's cargo gear is that this gear is useless during the vessel's voyage and constitutes a dead loss to its earning capacity. Thus any gear fitted should either be as light as strength will allow and/or be capable of the fullest utilization. The obvious way to fulfil this latter requirement is evident in the case of a mast upon which derricks serving two hatches are slung. The derricks of both hatches should be capable of serving either hatch. To observe this principle the author Dr. Heinrich Meyer has produced a new form of crosstree to the underside of which the topping blocks are attached. The height of the crosstrees above the derrick tables is sufficient to allow all derricks to pass vertically under the crosstrees without the necessity of unslinging. Alternative arrangements from two to six derricks and winches are described and more particularly the outfit of the Norddeutscher Lloyd steamer *Minden*, built by the Vulcan Werke of Stettin is dealt with. This vessel carries 6,400 tons deadweight and its four main hatches are served by four derricks slung on the fore and main masts respectively. Very complete working drawings of the crosstrees are given as well as a general arrangement and full description of the method of swinging the derricks under the cross trees. The system is being embodied in subsequent Norddeutscher Lloyd vessels and is also being adopted by the Kosmos Line.—*Dr. Heinrich Meyer, Werft, Reederei, Hafen*, April 22, 1923, p. 185; 3 pp.; 12ff.

The Stability of Ships.

This article, which refers exclusively to the case of the stability of vessels having a fluid cargo or a damaged compartment open to the sea forms a very comprehensive treatment of the subject. The solution of the most general case of a ship rolling in the midst of the usual analytically indefinable sea waves is dismissed as being equally beyond the scope of present-day mathematics and practical requirements. The simple case of a hanging weight is first dealt with and a very simple expression for the percentage loss of initial stability is deduced. The investigation is then extended to the case of a rolling weight and its analogy to the previous case is clearly demonstrated. The case of the fluid cargo is treated in a similar manner to that of the hanging and rolling weights. The importance of longitudinal subdivision in reducing the loss in initial stability is demonstrated and subsequently the most general case of the investigation of the changing initial stability during the filling of a tank is dealt with in a very clear and simple manner. Next the author extends his investigation to vessels in the damaged condition and shows that at least with merchant ships with long 'midship holds the effect of flooding these holds is to capsize the vessel. In this connection a very simple formula is given from which the safe length of 'midship hold may be determined in terms of the vessel's displacement, beam and intact metacentric height. The utility of longitudinal subdivision (*i.e.*, longitudinal bulkheads) is next discussed. It is pointed out that unless this subdivision is accompanied by fairly close transverse subdivision the system may be worse than useless. The case of end damage is briefly discussed as is also the stability of vessels being docked with slack ballast tanks. Finally, the utility of water chambers in annulling the rolling of vessels is touched upon. Generally, they are said to be unsatisfactory and if badly designed or in a particular seaway they may accentuate rather than annul the vessel's rolling. An exception is made in the case of the Frahm rolling tanks, whose design apparently leaves nothing to be desired. Finally, the author concludes that generally "the presence of moving water upon a rolling vessel is a grave danger. To undertake, in a seaway, the transference of a large quantity of water in a vessel is a serious mistake and an unjustifiable imprudence."—*Charles Doyère, Bulletin Techn. du Bureau Veritas*, February, 1923, p. 25, April, 1923, p. 81; 12 pp.; 28ff.

The T.S. Intermediate Liners "Antonio Delfino" and "Cap Norte."

The *Antonio Delfino* and the *Cap Norte*, forming the two latest additions to the Hamburg South America Line are of considerable interest. Built in 1921-22 by the Vulcan Werken of Hamburg and Stettin, they have the following general particulars: length over all 160·40 m., length b.p. 152·25 m., Breadth extr. 19·51 m., depth to bulkhead deck 12·80 m., draught (bulkhead) 8·03 m., gross tonnage 13,615, deadweight 9,650, speed 14 knots. Provision is made for 181 first-class passengers, 296 third-class cabin passengers and 1,050 third-class lower deck passengers. The construction of the vessels is described in great detail, plans of midship section, rudder stern post and shaft brackets, general arrangement complete with all decks, steering machinery, winches and finally photographs of the principal rooms being given.

The machinery arrangement is described in similar detail. The main machinery consists of two 2,900 I.H.P. triple-expansion engines having cylinders 680, 1,100 and 1,800 mm. diameter, piston stroke 1,300 mm., and working at 80 r.p.m. Five single-ended cylindrical oil-fired boilers of length 3,666 mm. and diameter 4,974 mm. are fitted. General arrangement, plans of engine and boiler rooms, engine arrangement, shafting and thrust block (multi-collar) propeller, oil burning arrangements, filters, pre-heaters and pumps are included in the article. It is interesting, in view of the present controversy with regard to the subdivision of passenger ships, to note that these vessels are subdivided according to the recommendations of the Bulkhead Convention. Further, that they are fitted with 23 life-boats, one motor boat, wireless telegraphy, submarine sound signalling apparatus (Echo-lot) and gyro-compass with automatic steering attachment. These facts speak well for the desire of the owners to ensure the safety of their ships.

Finally, so far as designers are concerned this article should form a valuable addition to the information on the Thrust Trials and model experiments of the *Antonio Delfino* recently given by Dr. Bauer before the Schiffbautechnischen Gesellschaft and reproduced in our contemporary the *Shipbuilding and Shipping Record* for June 7, 1923.—*Herrn. Dittmer and Müller, Werft, Reederei, Hafen*, June 7, 1923, p. 283; 20 pp.; 37 ff.

Systematic Research on Air Screws.

This article forms the conclusion of the series on the subject and contains much interesting matter. The American research by Durand and Leslie is confirmed in practically all respects by the author's own experiments. For given revolutions, low thrusts, and unrestricted diameter both researches confirm that a two-bladed

screw is about 6 to 7 per cent. more efficient than three or four-bladed. These latter, of course, however, admit of a 5 per cent. and 10 per cent. smaller diameter respectively being used to produce the same thrust. Contrary turning, two-bladed screws are not so efficient as the individual screws, but, like four-bladed screws, admit of a 10 per cent. smaller diameter being used for the same thrust. They are 3 per cent. better than three or four-bladed screws. On the influence of blade width ratio the German experiments show that the 10 per cent. blade width ratio was most efficient for low thrusts whilst the 12½ per cent. was most efficient for high thrusts. The best separation for contrary turning screws was found to be ·64 of the diameter of the screws. The experiments on tandem screws showed such a falling off in efficiency that as they could not receive practical application the results are not communicated. The comparison previously referred to of contrary turning screws was made on the basis of revs. fixed and diameter free. If the basis of diameter fixed and revs. free is adapted, then the contrary turning combination is as efficient as the single screw. For very high thrusts on the same basis, contrary turning screws are 4 per cent. more efficient than the single screws.—*Dr. K. Schaffran, Schiffbau,* April 18, 1923, p. 459; 5 pp.

The Container Transport Barge.

Germany has always paid great attention to the development of her canals and the economic combination of canal traffic and railway traffic has demanded the closest attention. The class of goods usually carried on canals is of the raw material type whose speed of transport is not of great importance. To deal with this traffic many interesting but costly proposals have been adopted but the stages in the transport from consumer to producer have usually necessitated many intermediate loadings and dischargings. The American container car idea eliminated a great deal of this intermediate manœuvring and the development of this idea to canal transport forms the basis of the article under review.

The necessary strength and watertightness which containers capable of carrying five tons should possess, absolutely prohibits, because of their weight, their embodiment in a canal barge of the usual design. However, the fact that the containers themselves could provide their own buoyancy if desired has led to the design of a most interesting type of barge. The after and forward ends of the vessel are of normal form and construction and are watertight. The parallel 'midship portion has normal side plating, deck stringer and bilge plating but otherwise the longitudinal, local and transverse strength is provided by a net work of girders which serve as platforms for the containers. The bottom plating is purposely made thin and is perforated to admit water to support the containers. The justification for the thin bottom plating is based upon the opinion that material a considerable distance away from the sides of the ship cannot usefully contribute to its longitudinal strength. Because of the perforated bottom the vessels draught does not greatly change between the light and the load condition. Such barges are no more difficult to tow than the ordinary type and in the light condition steer much better. Containers are brought alongside the canal by the train and are loaded by cranes into the barge. This method of transport is cheaper than rail transport but dearer than ordinary canal transport. When, however, the extra handling and likelihood of damage with the latter is considered, it is quite possible, when the transport of a certain class of goods is gone into, that a good case will be made for this new method of transport which thus combines convenience, safety and reduced costs.—*H. Schlatter, Schiffbau,* June 6, 13, 1923, p. 571; 4 pp.; 4ff.

Forced Draught for Tug Boilers.

The canal tugs in Germany can hardly be said to be ideally situated from the point of view of the attainment of the highest heat economy. Their constant stoppages and the necessity of collapsible funnels have made poor boiler performance up to the present almost inevitable. The natural draught rarely exceeded 2 to 3 mm. of water, and thus boilers could rarely produce their designed horse-power. However, Messrs. Has of Hamburg took up the problem, and finally produced a small turbo-blower which placed in the funnel uptake and driven by exhaust steam increased the draught up to 15 mm. of water. The more complete combustion of the fuel produced no less than 50 per cent. improvement in power and reduced the coal consumption by 15 per cent. This performance was made over two years ago, but all subsequent installations have confirmed the improvement. Tugs fitted with these turbo-blowers have no difficulty in quickly raising and maintaining their steam pressure and whereas coal consumptions of 1·05 to 1·15 kg. per horse-power hour were considered inevitable, consumptions as low as ·754 kg. have been obtained. Better combustion of course, carries with it further advantages. These are such as the use of untrained personnel, less cleaning of furnace, less smoke, longer life for fire bars, and smaller and hence non-collapsible funnel, all of which advantages are very real in inland navigation work.—*Baurat Ebelt, Werft, Reederei, Hafen,* May 22, 1923, p. 255; 3 pp.; 1ff.

THE LUBRICATION OF PROPELLER SHAFTS OF HOPPER BARGES.—(*Continued from page* 261.)

employed. The tail shafts, 8 in. in diameter, are without liners, but they are swelled at the ends in way of the bearings to 9 in. in diameter, as will be seen, and the bushes of the stern tubes are 4 ft. long at the outer and 2 ft. long at the inner ends.

The projecting device consists of a rubber ring, 1⅛ inch. thick, which is secured to and revolves with the propeller in a groove formed in a cast iron ring, bolted to the end of the stern tube. The pressure of oil from a tank placed in the engine-room casing forces the rubber ring against the brass bush in the casting and effectively prevents the escape of oil and the ingress of sand. Air and draw-off pipes are provided for the circulation of the oil and a steel shield is fitted to protect the rubber ring. The whole of the shaft from the propeller-nut to the gland in the engine-room is under oil pressure and so effectively protected and lubricated that on a propeller shaft recently drawn after 10 years' service, when the vessel had run a distance of 160,000 miles, no signs of deterioration or wear could be observed, in fact, the tool marks of the machining can still be seen plainly on the bearing parts of the shafts and bushes, as the photograph shows.

As a comparison with similar size of shafts employed on the same duty, having phosphor bronze liners, 5 ft. 6 in. long, at the outer end and ⅜ in. thick, with lignum vitæ bushes, 4 ft. long, it is interesting to note that these have a wearing period of about two years, and then the liner requires to be turned down to half its original thickness and the wood in the bush renewed. Of the four twin-screw barges built in 1911, only two shafts have been withdrawn and these were for inspection purposes. The rubber rings are composed of special composition to resist the softening action of the lubricating oils and are renewed, on an average, after being in use for four years. A compound oil is used as the lubricant which has the property of producing a creamy lather when mixed with water, and the free acid in a sample, after analysis, was found to be only 1·3 per cent., although the shaft had not been drawn or the stern tube drained for a period of 6½ years. The amount of oil used to keep the stern tubes filled is very moderate, taking on an average 10 gallons per month for each vessel at a cost of 2s. per gallon which is a very small charge in comparison with the cost of renewing liners and bushes of hopper barges with unprotected shafts.

SS. "MINNEWASKA"—MAIN PROPELLING MACHINERY.

We illustrate on page 271 some features of the turbine gear wheels of the new Atlantic Transport Liner *Minnewaska,* now completing at the Belfast yard of Harland & Wolff Ltd. This vessel is to re-commence the passenger service of the Atlantic Transport Line, which has been suspended since the end of the war, and her point of departure on this side will be the King George V Dock, London. The vessel's machinery consists of twin sets of single-reduction Brown-Curtis geared turbines. Each set comprises a high-pressure turbine which exhausts to a low-pressure turbine and thence to the condenser; the power developed is about 9,000 s.h.p., making a total for the ship of 18,000 s.h.p. An astern turbine is incorporated in each low-pressure casing.

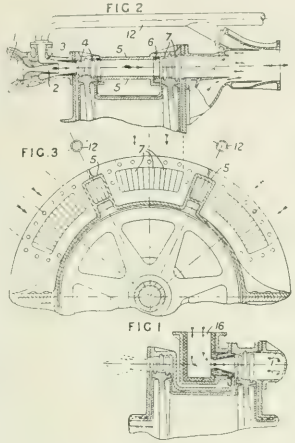
RECENT PATENTS RELATING TO MARINE ENGINEERING.

Compiled by E. P. Alexander & Sons, Chartered Patent Agents, 306, High Holborn, London, W.C.

No. 196,452. Turbines.

Hepburn, H. A., 33, *Kings Road, Haymills, Birmingham, March* 15, 1922. *No.* 7533. [*Class* 110 (iii).]

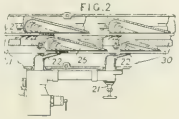
In an internal-combustion product turbine of the kind described in Specification 182,893 the operating gas passes through a set of rotor blades and then through a further set of blades, through which also at different sectors thereof induced gas passes, the two gases combining in the exhaust to be dealt with as described in Specification 182,893. The operating gas passes from a combustion chamber 1, Fig. 2, through ejector nozzles 2, rotor blades 4, passage 5, guide blades 6, rotor blades 7, to exhaust. A connection 12 from the vacuum space 3 of the ejector nozzles 2 to the exhaust ensures low pressure at the exhaust. The induced gas enters through nozzle boxes 16, Fig. 1, intermediate between the connection 5 as shown in Fig. 3, and passes through rotor blades 7 to the exhaust. The velocities of the induced and operating gas are arranged to be approximately equal when they enter the blades 7, and when they mix in the exhaust. Fixed guide blades may be provided at the entrance to the connecting passages 5. Steam may be used instead of or in addition to the operating gas. The clearance spaces between the nozzle boxes 16 and the recesses in the turbine casing in which they are located may be flushed out with cold air or filled with any desired substance.



No. 196,464. Turbines.

Wait, H. H., 907, *Jackson Street, Anderson, Indiana, U.S.A., March* 29, 1922, *No.* 9065. [*Class* 110 (iii).]

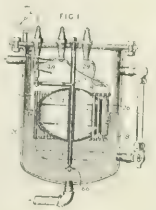
Circular-flow type; regulating.—In a multi-stage turbine of the circular-flow type, wherein the steam is returned to the same wheel by return guides, these guides 25 are located in apertures in the inter-stage diaphragm 30 between the two wheels 11, 12. The nozzles 22, which may be of different sizes for convenience in regulating, are separately controlled by hand-valves 21.



No. 196,499. Steam-generators.

Cohen, H. E., 144, *Renwick Place, Syracuse, New York U.S.A., May* 22, 1922. *No.* 14389. [*Classes* 37, 39 (iii), *and* 123 (ii).]

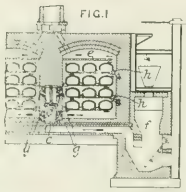
Steam is generated by passing an electric current through one or more open ended carbon tubes or the like supported by a float in the boiler water in a partially immersed position. The vertical carbon tubes 17, Fig. 1, are connected in series by end connectors 18 and are secured tightly against a rubber covering on the sides of a float 23 by a band 26 of insulating material. A switch 37, Fig. 2, in the current supply lines 39, 41 is automatically opened when the boiler water falls below a certain level. When the water falls, the float rests upon a collar 66 on a rod 62 inside the tubular guide member 29 and moves the rod downwards, thus opening a switch 48 in a branch circuit containing an electromagnet 44 acting on the locking arm 49 of the switch 37. The switch is opened by a spring 42 when the locking arm is released.



No. 196,513. Retort furnaces.

Tooley, C. P., 45, *Stapleton Street, New Wortley, Leeds, and Tooley, G. W.*, 21, *Woodlands Road, Cheetham Hill, Manchester, June* 15, 1922. *No.* 16621. [*Classes* 51 (i) *and* 51 (ii).]

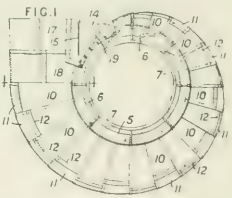
In a setting for horizontal or inclined gas retorts comprising superposed tiers of retorts arranged to provide a zig-zag series of flues for the heating-gases, gas and air are supplied to alternate flues through ports *d*, *g* at one side of the setting, and air is supplied to the intermediate flues through ports *h* at the other side. The gas inlet chamber *c* fed from the producer *f* serves two settings, as shown, and when a number of settings are arranged in line, the producers may be at the ends. A baffle *e* ensures that the gases keep close to the uppermost row of retorts.

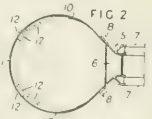


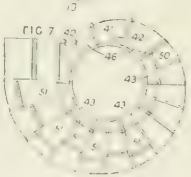
No. 196,590. Turbines.

Taylor, H. B., West Sunset Avenue, Chestnut Hill, Pennsylvania, U.S.A., March 14, 1923. *No.* 7435. *Convention date, April* 24, 1922. [*Class* 110 (iii).]

Casings, stators, working fluid supply.—A spiral casing for a turbine has a series of guide vanes 6 included in its central structure of cast metal and the surrounding volute casing is composed partly of cast metal and partly of plate metal. The central ring 5 with its vane 6, may be cast whole or in four or more sections; it has flanges 7 for attachment to the central structure supporting the adjustable guide vanes, and has flanges 8 for the attachment of the plate steel sections 10 of the casing. These sections 10 are flat at their inner ends and curved on substantially circular arcs for the outer plate sections 11 to complete the casing. The ends of the plates 10, 11 are curved or bulged slightly in the direction of the spiral to overlap the plates of the next section. The joints 12 between the plates 10, 11 of the different sections are staggered. The throat section 14 is cast and has a curved body 15 flanged for connection to the plate sections 17, 18, while the other side of the section 14 forms the body 19 terminating the spiral. The inner arc of the section is bolted to the vane ring 5. In a modification, the small portions of the volute are formed by cast sections instead of plate sections. The large entrance sections may also be cast so that a line of cast sections may be formed consisting of the large sections, the throat section and the small sections. In the modification shown in Fig. 7, the inner portions of the throat section 40 and of the small volute sections 41, 42 form part of the vane ring. The remainder of the ring is formed by the series of inner cast sections 43 each of which carries a vane 46. The plate sections 50, 51 are attached to flanges on the sections 43.







No. 196,913. Turbines.

Akt.-Ges. Brown, Boveri et Cie, Baden, Switzerland. April 16, 1923, *No.* 10332. *Convention date, April* 26, 1922. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act. Drawings to Specification.* [*Class* 110 (iii).]

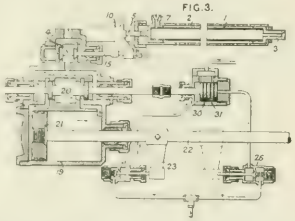
Working-fluid supply. In a steam turbine working with a high degree of superheat (steam temperature above 350° C.) that part of the turbine which works with highly superheated steam (above 300° C.) has only rotors running at small peripheral speed (below 160 m. per second). This peripheral speed is based upon the strength of the rotor material at the temperature determined by the utilised heat drop. The first stage with the small rotors may be an independent turbine driving its own dynamo, or it may be coupled through gearing to the main turbine, or it may be mounted on the same shaft as the main turbine.

No. 196,898. Steam-boilers.

Espujols, R. L. de, 8, *Promenade des Anglais, La Varenne St. Hilaire, Seine, France. March* 23, 1923. *No.* 8359. *Convention date, April* 28, 1922. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act.* [*Classes* 102 (i), 122 (ii), *and* 123 (ii).]

Feed-water, supplying and controlling.—The supply of steam or other motive fluid to the engine of a boiler feed-pump is controlled by the expansion and contraction of a tube or an expansible container so placed in communication with the boiler that it is filled with water or steam as the water level rises or falls. A horizontal tube 1, Fig. 3, connected to the boiler steam and water spaces through nipples 7, 3 is secured at one end in a fixed casing 2, and at the other end is formed with a projection 5 engaging with an arm 10, which, when the tube expands, is pressed against the spindle 15 of the valve 14, controlling the supply of steam to the engine cylinder 19. The engine slide valve 20 is operated by a piston 30 which is oscillated by the pressure of steam admitted to the opposite ends of its cylinder 31 in turn at the ends of the engine stroke. Valves 26 controlling the admission of steam to the cylinder 31 are opened in turn by an arm 23 secured to the piston rod 22. The arm presses against the end of the valve rod 15 through a projecting pin secured to a sliding disc in a cylinder on the end of the arm. The disc is acted upon in one direction by a spring and in the other direction by the pressure of boiler steam admitted to a flexible chamber through a nipple.

FIG. 3.



No. 196,902. Turbines.

Akt.-Ges. Brown, Boveri et Cie, Baden, Switzerland. April 4, 1923, *No.* 9263. *Convention date, April* 26, 1922. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act.* [*Class* 110 (iii).]

Casings; nozzles.—Quadrantal or semicircular nozzle rings *g* are secured between the main turbine casing *a* and a cover *b* so as to enable each ring to be inserted in one piece. In the form shown the joint *c* between the cover and the casing is in the central plane of the rings. The rings can be supplied with steam by separate pipes *f*, as shown in Fig. 1.

FIG. 1.

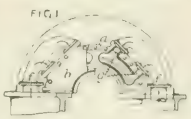
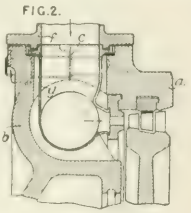


FIG. 2.



No. 196,931. Turbines.

Dufour, L., 6, *Route de Florissant, Geneva. April* 26, 1923, *No.* 11342. *Convention date, April* 28, 1922. *Not yet accepted. Abridged as open to inspection under Section* 91 *of the Act.* [*Classes* 110 (i) *and* 110 (iii).]

Axial-flow type; cooling; working-fluid supply.—A combustion turbine has several pressure stages and wheels which serve as turbine wheels and compressor wheels; these are so arranged that they impart to the gases a portion only of the pressure necessary for the working of the turbine, the remainder being imparted by an auxiliary centrifugal turbo-compressor. The turbine is shown with five wheels b^1 . . b^5 provided with ribs or blades *f* which project beyond the circumference of the wheels. At the top of the machine the projecting parts of the blades pass into the chambers k^1 . . k^5 through which the gases pass from the combustion chamber *n* and the tube *x*, and in these chambers the blades act as turbine blades. The lower part of the machine, comprising more than half its circumference, forms a compressor. The gases from chambers h^1 . . h^5 pass between fixed guide vanes *q* and enter the impeller; after leaving the blades *f* they pass through the diffusers *g* and chambers l^1 . . l^5 to pipes y^1 . . y^5 leading to sections of the auxiliary compressor. Twelve impeller wheels r . . r^{12} are mounted on the auxiliary compressor shaft a^2, Fig. 3, surrounded by diffusers g^1 . . g^{12}. The impellers r^1, r^2 act in series and discharge through a pipe z^4 into the atmosphere. The remaining impellers r^3 . . r^{12} act in series and the compressor part of the turbine wheel b^1 is included in the series between the impellers r^9, r^{10}. The compressor part of the turbine wheel b^5 acts in parallel with the impeller r^1, and the wheels b^2, b^3, b^4 act in parallel with the impellers r^7, r^5, r^3, respectively. The passage of the gases cools the turbine blades. During working air enters the pipe w^1, passes through the compressor impellers r^3 . . r^{12} and the associated turbine wheels b^4 . . b^1, passes through the pipe w^2 to the regenerator R^1, where it is heated by the exhaust gases from the turbine, and then passes through the pipe w^3 to the combustion chamber *n* to maintain the combustion of the crude petroleum which enters the chamber through the pipe v^5. The gases of combustion pass through the guide blades m^1 . . m^5 and chambers k^1 . . k^5 operating the turbine wheels b^1 . . b^5 as they pass; the last wheel b^5 is operated at a pressure below atmospheric pressure. The hot exhaust gases are conveyed by the pipe z^1 to the regenerator R^1 to heat the incoming gas, thence by the pipe z^2 to the refrigerator R^2, where they are cooled by circulating water and finally to the discharge pipe z^4 through the impeller r^1 and the compressor part of the turbine wheel b^5 arranged in parallel. Refrigerators may be placed between the stages or groups of stages of the auxiliary compressor and the compressor may be driven by the turbine. The number of turbine wheels placed in series and in parallel with compressor impellers may be varied. Specifications 181,377 and 188,322 are referred to.

FIG. 1

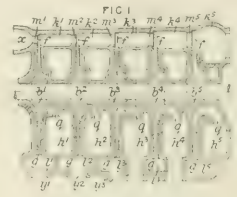


FIG. 2

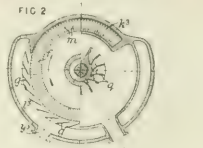
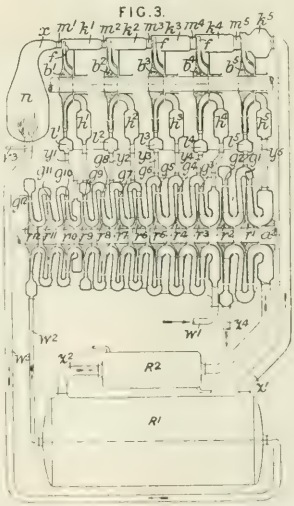


FIG. 3.



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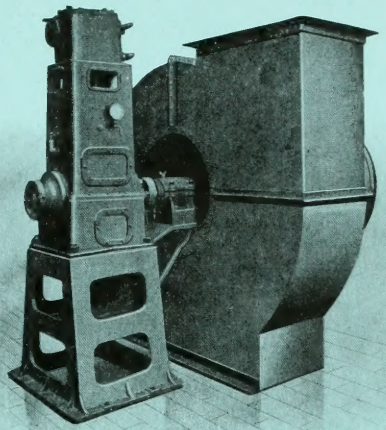

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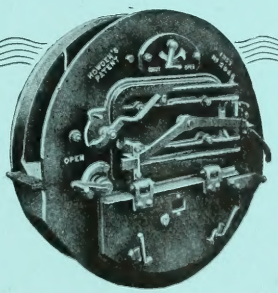

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